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ENERGY
COMMISSION

Large Commercial Building Distribution Systems

TECHNICAL REPORT

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Gray Davis, *Governor*

CALIFORNIA ENERGY COMMISSION

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Acknowledgements

In a program of this magnitude there are many people who contributed to its success. We owe the many staff members, faculty, and students of the different institutions our thanks for the superb work and long hours they contributed. All of their names may not appear in this report, but their efforts are visible in the many papers, reports, presentations, and thesis that were the major output of this program.

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Finally, we acknowledge the support and contributions of the PIER Contract Manager, Martha Brook, and the Buildings Program team under the leadership of Nancy Jenkins.

Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The Program's final report and its attachments are intended to provide a complete record of the objectives, methods, findings and accomplishments of the High Performance Commercial Building Systems (HPCBS) Program. This Commercial Building Energy Benchmarking attachment provides supplemental information to the final report (Commission publication # 500-03-097-A2). The reports, and particularly the attachments, are highly applicable to architects, designers, contractors, building owners and operators, manufacturers, researchers, and the energy efficiency community.

This document is the eighth of 22 technical attachments to the final report, and consists of research reports:

- Proposed Revisions to 2005 Title 24 Energy Efficiency Standards: Addition of HVAC Transport Efficiency Concept (E4P2.2T4a)
- Duct Thermal Performance Models for Large Commercial Buildings, for 2008 Title 24 Standard (E4P2.2T1)
- Benefits of Reducing Duct Leakage in Large Commercial Buildings (E4P2.2T2)
- Code Change Proposal for Duct Sealing in Large Commercial Buildings, for 2008 Title 24 Standard (E4P2.2T3)

The Buildings Program Area within the Public Interest Energy Research (PIER) Program produced this document as part of a multi-project programmatic contract (#400-99-012). The Buildings Program includes new and existing buildings in both the residential and the nonresidential sectors. The program seeks to decrease building energy use through research that will develop or improve energy-efficient technologies, strategies, tools, and building performance evaluation methods.

For the final report, other attachments or reports produced within this contract, or to obtain more information on the PIER Program, please visit http://www.energy.ca.gov/pier/buildings or contact the Commission's Publications Unit at 916-654-5200. The reports and attachments are also available at the HPCBS website: http://buildings.lbl.gov/hpcbs/.

Abstracts

Proposed Revisions to 2005 Title 24 Energy Efficiency Standards: Addition of HVAC Transport Efficiency Concept.

This memorandum report describes the ACM change proposed for the 2005 Title 24 Standards. The reporting change outlined in this report involves a new metric to address HVAC distribution system efficiency in large commercial buildings.

Duct Thermal Performance Models for Large Commercial Buildings.

This report reviews duct system modeling approaches and recommends an approach for benefits analyses in support of the 2008 Standards, as well as an approach that could be used by designers and for longer-term development of the Title 24 Standards. A significant element of this report is the publication of duct system modeling algorithms, embodied in the form of internally documented FORTRAN code. In the future, these algorithms could be added to simulation programs such as EnergyPlus.

Duct Leakage Impacts on VAV System Performance in California Large Commercial Buildings.

This report describes our assessment of the thermal performance impacts of improving duct systems in large commercial buildings, based on predictions obtained using the near-term simulation approach identified in the model review report.

Code Change Proposal for Duct Sealing in Large Commercial Buildings for the 2008 Title 24 Standard

There are not proposals included here because there are several substantial issues that must be addressed and resolved before such a proposal can be prepared. When the issues are resolved, the proposed changes will be posted on the HPCBS website.

California Energy Commission
Public Interest Energy Research Program

HPCBS # E4P2.2T4a

HPCBS

High Performance Commercial Building Systems

ACM-Change Proposal for 2005 Title 24 Update

Element 4. Low Energy Cooling
Project 2.2 - Efficient Distribution Systems

Mark Modera, Craig Wray
Lawrence Berkeley National Laboratory

October, 2002

Acknowledgement

This work was supported by the California Energy Commission, Public Interest Energy Research Program, under Contract No. 400-99-012 and by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098.

Ernest Orlando Lawrence Berkeley National Laboratory
One Cyclotron Road, Berkeley, California 94720

October 31, 2002

Bill Pennington
California Energy Commission
1516 9th Street MS 42
Sacramento, CA 95814

Dear Bill:

As we have discussed in the past, LBNL would like to get the concept of overall HVAC Transport Efficiency introduced into the 2005 standards process. Note that we are not proposing any modifications to the standard, nor to prescriptive or mandatory measures, nor to the modeling algorithms in compliance software. The proposal outlined in the attached document is simply a set of reporting changes, the goal of which is to provide feedback within the compliance process, and thereby within the design process, on the fraction of HVAC energy use that is going into blowing and pumping thermal energy and ventilation air around the proposed building. All of the variables required to calculate the defined HVAC Transport Efficiency are available within the standard reports from DOE-2, and therefore should not precipitate significant efforts on the part of ACM providers.

This proposal represents a first step in the transfer of LBNL's PIER research efforts on thermal energy distribution in large commercial buildings. We expect to provide better tools for evaluating HVAC Transport Efficiency, as well as proposals for how to improve that efficiency, in future standards revision processes.

Thank you in advance for your thoughtful consideration of our proposal, and please do not hesitate to contact me with any feedback or questions.

Sincerely,

Mark Modera
Staff Scientist

Cc: Craig Wray (LBNL), Max Sherman (LBNL)

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Proposed Revisions to 2005 Title-24 Energy Efficiency Standards: Addition of HVAC Transport Efficiency Concept

Prepared by:
Mark Modera

NON-RES MANUAL

Section 4.1.2 Basic Mechanical Concepts

A. Definitions of Efficiency

HVAC Transport Efficiency: the ratio between the energy expended to transport heating, cooling and ventilation throughout the building, and the total thermal energy delivered to the various zones in the building. The transport energy includes all distribution-fan, ventilation-fan and pump consumption (excluding DHW pumps), and the thermal energy delivered is the sum of all zone loads. This ratio can be calculated both over the course of the year, and under design conditions.

TE = (distribution fan energy + ventilation fan energy + non-DHW pump energy)/(total thermal load)

ALTERNATIVE CALCULATION METHOD

Should include the following section:

2.4.2.36 HVAC Transport Efficiency

Description: ACMs shall be able calculate the ratio between the energy expended to transport heating, cooling and ventilation throughout the building, and the total thermal energy delivered to the various zones in the building.

Modeling Rules: The transport energy includes all distribution-fan, ventilation-fan and non-DHW pump consumption, and the thermal energy delivered is the sum of all zone loads. This ratio must be calculated both over the course of the year, and under design conditions.

TE = (distribution fan energy + ventilation fan energy + non-DHW pump energy)/(total thermal load)

COMPLIANCE FORMS

MECH-4

Should be modified to report the design HVAC Transport Efficiency at the bottom using the following format:

HVAC Transport Efficiency = (Total Column F + Total Design non-DHW Pumping Power)/ (Total Heating +Total Cooling (from section 2.))

Would also be desirable to report the annual average value of TE, however some additional values would also have to be calculated and reported on the MECH-4 form (e.g. annual pump energy and fan energy, and total annual loads). All of these values should be available from standard DOE-2 reports.

California Energy Commission
Public Interest Energy Research Program

LBNL-53410
HPCBS # E4P22T1

HPCBS

High Performance Commercial Building Systems

Duct Thermal Performance Models

Element 4. Low Energy Cooling
Project 2.2 - Efficient Distribution Systems

Craig Wray
Lawrence Berkeley National Laboratory
October, 2003

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This work was supported by the California Energy Commission, Public Interest Energy Research Program, under Contract No. 400-99-012 and by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program of the U.S. Department of Energy under Contract No. DE-AC03-76SF00098.

LBNL-53410

Duct Thermal Performance Models for Large Commercial Buildings

Craig Wray

Environmental Energy Technologies Division
Indoor Environment Department
Lawrence Berkeley National Laboratory
Berkeley, CA 94720

October 2003

This report describes work supported by the California Energy Commission through the Public Interest Energy Research program under contract no. 400-99-012-1, and by the Assistant Secretary for Energy Efficiency and Renewable Energy, Building Technologies Program, of the U.S. Department of Energy under contract no. DE-AC03-76SF00098.

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LBNL staff members in the Building Technologies Department were also helpful and supportive of this work, particularly regarding the intricacies of DOE-2.1E and EnergyPlus. The author wishes to thank Phil Haves, Fred Winklemann, Fred Buhl, Joe Huang, and Steve Selkowitz.

The duct modeling discussions with Lixing Gu, Jim Cummings, Muthusamy Swami, and Don Shirey of the Florida Solar Energy Center and Pete Jacobs of the Architectural Energy Corporation helped define the approaches that we selected for duct thermal performance modeling in large commercial buildings, and the author is grateful for their input.

Advice from Michaël Kummert of the Solar Energy Laboratory, University of Wisconsin-Madison regarding TRNSYS compiling issues was greatly appreciated; as was the explanation by Tim McDowell of Thermal Energy System Specialists about the details involved in coupling TRNSYS and EnergyPlus.

This project evolved from the ideas and work of Mark Modera (LBNL), who was the original project lead. Even after he left the project, Mark continued to provide contributions and was a valuable resource for the project team.

Finally, the author would like to acknowledge the support and contributions of the PIER Contract Manager, Martha Brook, and the Buildings Program team under the leadership of Nancy Jenkins.

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EXECUTIVE SUMMARY

Introduction. Despite the potential for significant energy savings by reducing duct leakage or other thermal losses from duct systems in large commercial buildings, California Title 24 has no provisions to credit energy-efficient duct systems in these buildings. A substantial reason is the lack of readily available simulation tools to demonstrate the energy-saving benefits associated with efficient duct systems in large commercial buildings.

Purpose. The overall goal of the Efficient Distribution Systems (EDS) project within the PIER High Performance Commercial Building Systems Program is to bridge the gaps in current duct thermal performance modeling capabilities, and to expand our understanding of duct thermal performance in California large commercial buildings. As steps toward this goal, our strategy in the EDS project involves two parts: 1) developing a whole-building energy simulation approach for analyzing duct thermal performance in large commercial buildings, and 2) using the tool to identify the energy impacts of duct leakage in California large commercial buildings, in support of future recommendations to address duct performance in the Title 24 Energy Efficiency Standards for Nonresidential Buildings.

Project Objectives. The specific technical objectives for the EDS project were to:

1. Identify a near-term whole-building energy simulation approach that can be used in the impacts analysis task of this project (see Objective 3), with little or no modification. A secondary objective is to recommend how to proceed with long-term development of an improved compliance tool for Title 24 that addresses duct thermal performance.
2. Develop an Alternative Calculation Method (ACM) change proposal to include a new metric for thermal distribution system efficiency in the reporting requirements for the 2005 Title 24 Standards. The metric will facilitate future comparisons of different system types using a common “yardstick”.
3. Using the selected near-term simulation approach, assess the impacts of duct system improvements in California large commercial buildings, over a range of building vintages and climates. This assessment will provide a solid foundation for future efforts that address the energy efficiency of large commercial duct systems in Title 24.

This report describes our work to address Objective 1, which includes a review of past modeling efforts related to duct thermal performance, and recommends near- and long-term modeling approaches for analyzing duct thermal performance in large commercial buildings. Modera (2002) and Wray and Matson (2003) respectively describe work to address Objectives 2 and 3.

Project Outcomes

<u>*Recommended Short-Term Modeling Approach*</u>. Our review of 188 documents related to past modeling efforts, and supplemental discussions with other simulation experts, has helped define a set of modeling principles that can be used to guide duct thermal performance modeling for large commercial buildings. Based on this review, we conclude that the best approach for our impacts analysis task is to build upon past research that used DOE-2 and TRNSYS in a sequential method to evaluate HVAC system performance.

An advantage of this approach is that DOE-2 prototypical models for a large commercial California building are available, as are TRNSYS component models that LBNL developed in the past to model duct leakage effects in VAV systems. Another advantage is that this modeling approach and its results for a California building have already been validated, and no substantial changes are required to the simulation tool to carry out our impacts analysis. No other whole-

building modeling approach to assess duct system performance for large commercial buildings is currently as advanced as this approach. To assist other modelers, this report presents the source code for the TRNSYS component models, which were never published.

Recommended Long-Term Modeling Approach. Although DOE-2.1E Version 110 is the reference simulation tool for Title 24 compliance evaluations, we have concluded that its duct modeling limitations, convoluted structure, and the lack of government support for future development make it unsuitable as a platform for long-term modeling of duct thermal performance in large commercial buildings. Instead, we have suggested that EnergyPlus, which is based in part on DOE-2, be developed to include component models like the TRNSYS ones that we identified for short-term use in our impacts analysis task. Currently, EnergyPlus has no duct performance models, but we expect that the recommended enhancements could be applied in a relatively straightforward manner. This approach has the advantage that EnergyPlus is better suited than DOE-2.1E for future analyses of innovative low-energy cooling designs.

Our recommendation carries with it a set of challenges that need to be met by the summer of 2005 if EnergyPlus is to be used in support of the 2008 Title 24 Standards: 1) an interface needs to be rapidly developed to facilitate program use in Title 24 compliance analyses, 2) duct performance models need to be integrated with the program, 3) EnergyPlus needs to be validated against measured data and certified as either an alternative or primary compliance analysis tool, and 4) utilities to convert DOE-2 input files for use in EnergyPlus are needed to help current DOE-2.1E users migrate to using EnergyPlus. Further collaborative efforts between DOE and the California Energy Commission would help ensure that these challenges can be met, and would likely lead to substantial energy reduction benefits in California over the long-term.

Recommendations for Further Work. Before duct performance in large commercial buildings can be accounted for in Title 24 nonresidential building energy standards, several issues must be addressed and resolved. These include:

1. Specifying reliable duct air leakage measurement techniques that can be practically applied in the large commercial building sector.
2. Defining the duct leakage condition for the standard building used in Title 24 compliance simulations.
3. Assuring consistency between simulated duct performance impacts and actual impacts.
4. Developing compliance tests for the Alternative Calculation Method (ACM) Approval Manual (CEC 2001b) to evaluate duct performance simulations.

Three additional steps will be required to further develop duct-modeling capabilities that address limitations in existing models and to initiate strong market activity related to duct system improvements. We recommend that these steps include:

1. Implementing duct models in user-friendly commercially-available software for building energy simulation, validating the implementations with case studies and demonstrations, and obtaining certification for software use as a primary or alternative compliance tool in support of the Title 24 Nonresidential Standards.
2. Developing methodologies to deal with airflows entering VAV boxes from ceiling return plenums (e.g., parallel fan-powered VAV boxes), to deal with duct surface heat transfer effects, and to deal with static pressure reset and supply air temperature reset strategies.
3. Transferring information to practitioners through publications, conferences, workshops, and other education programs.

1. INTRODUCTION

1.1 Background

Previous research suggests that duct systems in California commercial buildings suffer from a number of problems, such as thermal losses due to duct air leakage. For example, measurements by Diamond et al. (2003) in a large commercial building confirmed predictions by Franconi et al. (1998) that duct leakage can significantly increase HVAC system energy consumption: adding 15% duct leakage at operating conditions leads to a fan power increase of 25 to 35%. Diamond et al. also estimated that eliminating duct leakage airflows in half of California's existing large commercial buildings has the potential to save about 560 to 1,100 GWh annually ($60-$110 million per year or the equivalent consumption of 83,000 to 170,000 typical California houses), and about 100 to 200 MW in peak demand.

California Title 24, Part 6 (CEC 2001a) is one of the most advanced energy codes in the United States. The impacts of duct thermal performance in residences are already addressed by Title 24 compliance procedures; duct-system energy efficiency requirements have recently been added for small commercial buildings with individual packaged equipment serving 5,000 ft² or less where ducts are located in spaces between insulated ceilings and the roof, or outside the building; and new requirements for duct performance in other small commercial buildings are being developed. However, despite the potential for significant energy savings by reducing thermal losses from duct systems in large commercial buildings, Title 24 has no provisions to credit energy-efficient duct systems in these buildings. A substantial reason is the lack of readily available simulation tools to demonstrate the energy-saving benefits associated with efficient duct systems in large commercial buildings.

1.2 Project Objectives

The work reported here is part of the Efficient Distribution Systems (EDS) project within the PIER High Performance Commercial Building Systems Program. The EDS project goal is to bridge the gaps in duct system modeling capabilities, and to expand our understanding of duct thermal performance in California's large commercial buildings, by following through on the strategy outlined by Xu et al. (1999a). As steps toward this goal, the project involves three specific technical objectives:

1. Identify a near-term whole-building energy simulation approach that can be used in the impacts analysis task of this project (see Objective 3), with little or no modification. A secondary objective is to recommend how to proceed with long-term development of an improved compliance tool for Title 24 that addresses duct thermal performance.
2. Develop an Alternative Calculation Method (ACM) change proposal to include a new metric for thermal distribution system efficiency in the reporting requirements for the 2005 Title 24 Standards. The metric will facilitate future comparisons of different system types using a common "yardstick".
3. Using the selected near-term simulation approach, assess the impacts of duct system improvements in California large commercial buildings, over a range of building vintages and climates. This assessment will provide a solid foundation for future efforts that address the energy efficiency of large commercial duct systems in Title 24.

To meet these objectives, we carried out the EDS project in steps. Work to address Objectives 1 and 2 was carried out in parallel, and was followed by work on Objective 3. This report describes our efforts related to Objective 1:

- Carrying out a review of documents related to past HVAC system modeling efforts, supplemented by discussions on these issues with other simulation experts;
- Defining a set of modeling principles and publishing HVAC component models that can be used now to guide duct thermal performance modeling for large commercial buildings; and
- Identifying what aspects can be carried forward for use in Title 24 compliance processes related to large commercial buildings.

The efforts related to Objective 1 are necessary precursors to assessing the impacts of duct system improvements in California large commercial buildings, which is the focus of Objective 3. A follow-on report by Wray and Matson (2003) describes the latter effort in detail[1].

Regarding Objective 2, the California Energy Commission has accepted the ACM change that Modera (2002) proposed for the 2005 Title 24 Standards to address HVAC distribution system efficiency in large commercial buildings. The metric of interest, HVAC Transport Efficiency, characterizes the overall efficiency of the thermal distribution system as the ratio between the energy expended to transport heating, cooling, and ventilation throughout a building and the total thermal energy delivered to the various conditioned zones in the building. Because the ACM proposal is for a set of reporting changes, implementing the changes in existing Title 24 non-residential compliance software should not require significant effort from ACM providers.

This project contributes to the PIER program objective of improving the energy cost and value of California's electricity in two ways. One is by developing analytical methods to show that well designed duct systems in large commercial buildings can save much of the energy used to move and condition air. The other is by making progress toward new requirements for commercial duct system efficiency in future revisions of Title 24. We expect that the new analytical capabilities and performance requirements will ultimately result in smaller capacity, more energy-efficient building systems, which will also reduce peak electrical demand from California's commercial building sector and improve the reliability and quality of California's electricity.

1.3 Report Organization

In **Section 2, Modeling Context**, we discuss issues that delineate modeling needs, and that provide rationale for selecting near- and long- term modeling approaches.

In **Section 3, Modeling Reviews**, we present the key results from our review.

In **Section 4, Conclusions and Recommendations**, we present what we learned from the research and what we recommend for future activities.

Following the **Glossary**, there are two **Appendices**:

"**Appendix I.** Bibliography" lists the 188 documents related to HVAC air-handling system performance simulation and assessment that we reviewed in our search for information about duct thermal performance modeling.

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[1] Wray, C.P. and N.E. Matson. 2003. "*Duct Leakage Impacts on VAV System Performance in California Large Commercial Buildings*". Lawrence Berkeley National Laboratory Report. LBNL-53605.

Particularly relevant documents are:

- Past efforts that assessed DOE-2 capabilities for modeling ducts in commercial buildings (Modera et al. 1999, Xu et al. 1999a),
- A recent doctoral dissertation that used DOE-2 and TRNSYS to model duct thermal losses in large commercial buildings (Franconi 1999),
- Recent DOE-2.2 modeling efforts that assessed duct system improvements for small commercial buildings (PG&E 2003a, 2003b), and
- Discussions of residential duct thermal models (Gu et al. 1998a, 1998b), which are the basis for a recent proposal by the Florida Solar Energy Center to integrate duct thermal performance modeling into EnergyPlus for residential and small-commercial buildings.

To supplement these reviews, we also hosted several conference calls and meetings between LBNL and Florida Solar Energy Center staff to map out a long-term strategy for adding duct-modeling capabilities into EnergyPlus for large-commercial buildings.

"**Appendix II.** TRNSYS Duct Performance Subroutines" lists the FORTRAN source code implementations of the seven models that Franconi (1999) used in a DOE-2 / TRNSYS simulation approach to predict the effects of duct leakage on HVAC system performance.

2. MODELING CONTEXT

The background information in this section serves as a basis for evaluating and selecting methods that evaluate duct performance impacts. We first summarize the two different compliance paths in Title 24 to help the reader understand the reasons why one needs simulation tools to evaluate duct performance. Next, we describe duct system types that are common in California's large commercial buildings, present an example to illustrate the effects of duct system deficiencies, and describe the underlying principles that govern duct performance.

2.1 California Title 24 – Compliance Path Overview

Two compliance paths are available for non-residential buildings that are subject to California Title 24 requirements:

1. The simplest approach is *prescriptive*: compliance is achieved by designing and constructing the building to meet specified minimum characteristics for the envelope, space-conditioning system, hot-water heating system, and lighting system.
2. The more complex approach is *performance* based and is intended to provide flexibility for innovative design and construction by allowing efficiency tradeoffs between various building components. As part of this approach, energy uses of the proposed building and of a standard building in the same climate zone are calculated using a simulation tool. Compliance at the design stage is achieved if the energy use calculated for the proposed building does not exceed the energy use calculated for the standard building. The standard building is physically similar to the proposed building, but is equipped with components that meet the requirements outlined in the prescriptive compliance approach.

Only two software packages are commercially available and certified for performance-based compliance evaluations: Perform95 from the Commission, and EnergyPro from EnergySoft, LLC. Both programs serve as a front-end to DOE-2.1E.

DOE-2.1E is an hour-by-hour energy analysis program that calculates whole-building energy performance and life-cycle economics (Winkelmann et al. 1993). Other simulation programs could be used to evaluate compliance, but the Commission must certify each one for such use. To be considered for certification, the alternative tool must meet the analysis specifications outlined in the Alternative Calculation Method (ACM) Approval Manual (CEC 2001b) and must be tested to assess modeling accuracy relative to DOE-2.1E Version 110 predictions. Accuracy is assessed using 76 conformance tests that involve several building prototypes, climate zones, and design/system permutations; each test systematically varies one or more features that impact building energy use. Acceptable accuracy means that the performance differences between the proposed and standard buildings calculated using the alternative tool must be within 15% of the differences calculated using DOE-2.1E.

2.2 California Duct Systems

Using survey data collected from 1988 through 1993 by or for California utilities and for the California Energy Commission, Modera et al. (1999) determined that there are three basic types of duct systems in California commercial buildings:

- *Single-duct* systems generate either a cool or warm air stream at the air-handler. The supply air is delivered to the conditioned zones through a single duct system connected to the air-handler. Reheat coils at individual terminal units can be used to add heat to the supply air when needed.
- *Dual duct* systems generate a cool air stream and a warm air stream at the air-handler. Each air stream is supplied to terminal boxes through a separate duct system. The terminal boxes mix the air streams before the supply air enters the zones.
- *Multizone* duct systems also generate a cool air stream and a warm air stream at the air-handler, but they use dampers at the air-handler instead of at a terminal box to mix the cool and warm air streams for each zone. Each zone’s supply air is delivered through a separate duct system (this system is somewhat like several single-duct systems operating in parallel).

All of these duct systems use one of two methods to control the amount of energy supplied to each zone. A *constant-air-volume (CAV)* system delivers a fixed quantity of supply air to the conditioned space and maintains desired conditions by varying the temperature of the supply air. A *variable-air-volume (VAV)* system maintains space temperature by varying the quantity of supply air, generally at a fixed temperature.

Based on the floor area served by these duct systems (Modera et al. 1999), the most common system across different building types is the single duct CAV system (71%). The next most common system type is the multizone system (19%). Single-duct VAV systems (8%) and dual duct systems (2%) serve the remainder of the floor area. Note that the fraction of multizone systems might be overrepresented by these data. Modera et al. indicated that the survey data may include some inappropriate affirmative responses for multizone systems. In some cases, the respondent may have called a system that serves more than one zone a multizone system, even though the system is not really a multizone system as described above. For example, some of the multizone systems might actually be single-duct VAV systems that serve multiple zones.

The fractions of floor areas served by CAV and VAV system types are difficult to determine, because the fractions for multizone and dual-duct systems are unknown. However, based on data from Modera et al. (1999) and EIA (2002), the fraction of VAV systems may be in the range of 8 to 34%. The EIA data indicate that VAV systems serve 34% of the large commercial building floor area in the U.S. Pacific region, which includes California.

Although there are substantially fewer VAV systems than CAV systems in California, it is clear that VAV systems are used in a significant fraction of California buildings and need to be addressed when developing duct models for large commercial buildings. A reason to focus on VAV systems is that if one is able to model a VAV system, then a CAV system can also be modeled (it is a simplification of a VAV system). Another reason is that an EPRI study (Pietsch 1991) suggested a significant national trend over the past 30 years towards the use of VAV systems in new construction (e.g., about 75% of new duct systems in the period 1980 through 1990 were VAV systems).

Of the floor area served by single-duct VAV systems, the data from Modera et al. (1999) indicate that most (98%) of it is in large office buildings; the remainder (2%) is primarily in hotel and retail buildings. For this reason, we focused on large office buildings in our study.

2.3 Effects of Duct Deficiencies

In large commercial buildings, duct systems and the effects of deficiencies in these systems are much more complex than in most residential and small-commercial buildings. As an example to illustrate the effects of duct system deficiencies, consider a large commercial building equipped with a single-duct terminal-reheat VAV system that has leaky supply ducts located within a ceiling return air plenum.

When conditioned air leaks from the supply ducts, the heating or cooling energy associated with leakage heats or cools the return air and changes its temperature (and enthalpy). Depending on the temperature difference across each surface that separates the plenum from adjacent conditioned spaces and the outdoors, some of the energy associated with the leakage airflow is transferred from the plenum by conduction across these surfaces. The energy transferred by conduction between the plenum and adjacent zones may be beneficial or detrimental to zone loads. For example, when there is simultaneous heating of perimeter zones and cooling of the core zone, the heating energy associated with leakage from ducts that serve the perimeter zones will tend to increase plenum temperatures; the cooling energy associated with leakage from ducts that serve the core zone will tend to decrease plenum temperatures. A net increase in plenum temperatures will increase the core-zone cooling load and decrease the perimeter-zone heating loads. Conversely, a net decrease in plenum temperatures will decrease the core-zone cooling load and increase the perimeter-zone heating loads.

If the VAV boxes deliberately induce airflows from the ceiling plenum (driven by induction effects or by VAV box fans), the change in return air enthalpy affects the mixed supply air enthalpy within and downstream of the VAV box. This in turn affects the energy that is transferred to the conditioned spaces by these airflows. It can also affect VAV box reheat coil loads (e.g., reduced return air enthalpy due to cool supply air leakage upstream of the VAV box or from other ducts reduces the VAV box mixed air enthalpy and increases reheat coil loads).

A change in return air temperature due to duct leakage will also change cooling coil loads when the economizer is not operating. For example, consider an air-handler with an economizer that is controlled based on dry-bulb temperatures (rather than on enthalpies). When the outdoor air temperature is above the return air temperature high-limit set point, the amount of outdoor air entering the air-handler is the minimum required for ventilation. The remainder of the mixed airflow entering the air-handler (same flow rate as the supply airflow) is return air. Mechanical cooling is used to maintain the desired supply air temperature. In this case, the change in return air enthalpy due to duct leakage will affect the mixed air enthalpy entering the air-handler coils, and therefore will affect the cooling coil loads (e.g., reduced return air enthalpy due to cool supply air leakage reduces mixed air enthalpy and therefore reduces cooling coil loads). To maintain the desired air pressure differentials across the building envelope, some return air is discharged outdoors. This arrangement means that some of the heating or cooling energy associated with leakage is discharged to outdoors and is not recaptured at the air-handler.

When the outdoor air temperature is between the desired supply air temperature and return air temperature high-limit set point, the economizer operates with 100% outdoor air and no return air enters the air-handler (all of the return air is discharged outdoors). In this case, even though mechanical cooling is used as a supplement to maintain the desired supply air temperature, the change in return air enthalpy due to duct leakage does not affect mixed air enthalpy or cooling coil loads. When the outdoor air temperature is below the desired supply air temperature, there is no mechanical cooling and duct leakage again has no impact on air-handler coil loads. However, to maintain the desired supply air temperature in this case, a change in return air temperature (e.g., due to duct leakage) will cause the economizer to alter the amounts of return air and outdoor air that enter the air-handler.

In the case of a VAV box with leaky downstream ducts, the duct leakage means that insufficient heating or cooling energy is delivered to the conditioned spaces. As a result, the thermostat call for heating or cooling is not satisfied and the thermostat calls for more air to be supplied through the VAV box. To deliver more supply air, the VAV box primary air damper opens further, which in turn reduces the resistance to airflow in the duct system. Consequently, to maintain the main duct static pressure at its set point, the supply fan airflow must increase to compensate for the downstream leakage airflows. Upstream leakage has a similar effect on supply fan airflow, but no effect on VAV box flows (unless the supply fan is too small to maintain duct static pressure in the leaky duct system).

Because the relationship between fan power and airflow is somewhere between a quadratic and cubic function (as described later in Section 2.4.2), the increase in supply airflow to compensate for duct leakage means that supply fan power consumption increases significantly, with a large fraction of this fan power used just to move the leaking air. Increasing the fan power also increases cooling coil loads when mechanical cooling is being used to maintain the desired supply air temperature (when the economizer is operating at 100% or minimum outdoor air). Specifically, the coil load increase occurs because the heat created by the increased fan power tends to increase the supply air temperature downstream of the fan. In response, the cooling coil water valve open furthers to provide more cooling to maintain the desired supply air temperature.

2.4 Duct System Performance Principles

A brief overview of duct air leakage, fan performance, and duct surface heat transfer principles is presented here, with a focus on supply ducts; Parker et al. (1993), Bourdouxhe et al. (1998), and ASHRAE (2001a; 2001b) provide more detailed descriptions. Return ducts are governed by similar principles.

2.4.1 Duct Air Leakage

A power law can be used to describe the relationship between the flow through the leaks in ducts and the static pressure in the duct relative to surrounding space:

$$Q_{leak} = C_1 \cdot \Delta p^{n}_{(duct-space)} \tag{1}$$

Equation 1 indicates that higher system static pressures lead to higher duct air leakage rates for a fixed "hole size" (characterized by the coefficient C_1 and the exponent n). For leaks that look like orifices (e.g., large holes), n is 0.5; for leaks with some length (e.g., lap joints between duct sections), n is larger (on the order of 0.6 or more).

When testing a duct section for leakage by fan pressurization, a measured pressure differential is applied to the test section through a fan that blows air from the surrounding space into the duct. The supplied flow that maintains this pressure differential is determined using a flow meter. By using several data points for Q and Δp, one can solve for C_1 and n using a least squares fit. The effective "hole size" characteristics determined by this pressurization test represent an aggregate of all the leaks in the test section. A common method of reporting the duct leakage uses the leakage class (CL) metric, which expresses the leakage flow in cfm at a reference pressure (1 in. of water, 250 Pa), normalized per 100 ft^2 of duct surface area.

Duct Leakage in CAV Systems. In CAV systems, the static pressure in the duct is typically not actively controlled: the static pressure at the fan exit is dependent on system flow resistance and fan performance characteristics.

The static pressures across duct leaks $\Delta p_{(duct\text{-}space)}$ can be related to the static pressure drop through the downstream section of the duct after the fan Δp_{duct}. Assuming a linear pressure drop through the duct, and that the zone supply air exits the diffuser and enters the space at ambient static pressure, the average static pressure in the duct equals about half the static pressure drop through the duct. If turbulent flow is assumed, the airflow rate through the duct Q_{duct} is related to the duct pressure drop according to the square law. This pressure-flow relationship can be expressed as:

$$\Delta p_{(duct-space)} = \frac{\Delta p_{duct}}{2} = C_2 \cdot \left(\frac{Q_{duct}^2}{2} \right) \tag{2}$$

If large holes are assumed in the ducts, then $n = 0.5$ in Equation 1. Assuming that the average duct static pressure corresponds with the average leakage rate, Equation 2 can be substituted into Equation 1 to solve for the average leakage rate as follows:

$$Q_{leak} = C_1 \cdot \Delta p_{(duct-space)}^{0.5} = C_3 \cdot Q_{duct} \tag{3}$$

where

$$C_3 = C_1 \cdot \left(\frac{C_2}{2} \right)^{0.5} \tag{4}$$

In this rough simplification, the fractional leakage ratio C_3 remains fixed regardless of system flow rate and fan pressure. This result assumes that:

- Duct airflow is turbulent,
- Duct pressure varies linearly along the length of the duct,
- Average duct static pressure indicates average leakage rate, and
- Duct leaks are large and have a pressure exponent of 0.5.

While these assumptions are plausible for CAV systems, they are not consistent with the conditions produced in some parts of a VAV system.

Duct Leakage in VAV Systems. In contrast to CAV systems, VAV systems maintain a constant static pressure at some point in the duct system upstream of the VAV boxes (except when static pressure reset control strategies are implemented). Consequently, duct air leakage occurring upstream of VAV boxes will have essentially the same flow rate at part-load fan operation as at design conditions. This means that the leakage fraction in the upstream duct sections is not constant; instead, it varies as the supply flow varies.

The duct static pressure downstream of a VAV box is influenced by the position of the VAV box dampers, as the damper modulates in response to the differential between the room temperature and the thermostat set point. Thus, the duct section downstream of a VAV box behaves much like a CAV system and downstream duct leakage occurs at a fixed fraction of the supply air entering that section.

2.4.2 Fan Performance

Fan electric power is dependent on the fan air power (product of the flow through the fan, and the total pressure rise across the fan), the blade efficiency, and the motor and drive efficiencies. The pressure rise across the fan must be sufficient to overcome the pressure drop in the system. This system pressure drop depends on the pressure drops across duct and duct-like elements (e.g., dampers, fittings), coils, and filters, as well as the static-pressure set point.

Duct and duct-like pressure drops increase as a function of the square of the flow through them. If ducts were the only component in the system, the fan air power would be a cubic function of the flow through the system. However, filters and coils usually follow a power-law functional relationship between pressure drop and flow. For these elements, the pressure drop is proportional to the flow raised to $1/n$:

$$\Delta P_{element} = \frac{Q_{element}^{1/n}}{C_{element}} \tag{5}$$

The value of n for the elements varies from 0.5 to 1. If one of these elements were the only one in the system, the fan power would be a function of the fan flow raised to the power $(1+1/n)$. This bounds the fan air power as somewhere between a square and a cubic function of fan flow.

Knowing the design flows and pressure drops (along with the appropriate n's), it is possible to plot the system pressure drops over a range of flows. If the system resistance varies due to changes in VAV box damper positions, the plot would consist of a family of system curves. Each system curve presents the pressure-drop/flow relationship for a fixed system resistance. When the system performance curves are plotted along with fan performance curves on flow versus pressure plots, the system-fan curve intersections define a locus of unique system operating points.

In many hourly simulation programs, including DOE-2, the fan performance subroutines are based on a third-order polynomial relating fractional fan shaft power to fan flow part load ratio (Brandemuehl et al. 1993). The form of the equation is:

$$FPR = c_0 + c_1 \cdot PLR + c_2 \cdot PLR^2 + c_3 \cdot PLR^3 \tag{6}$$

where

- FPR: Fan power ratio, which is the dimensionless ratio of the fan shaft power at a particular time to the fan shaft power under design conditions;
- PLR: Part load ratio, which is the dimensionless ratio of the fan flow at the same time to the fan flow under design conditions; and
- $c_0 \ldots c_3$: Constant coefficients for the curve fit. The specific coefficients depend on the pressure drop, pressure control, and flow characteristics of the system.

2.4.3 Duct Surface Heat Transfer

Heat transfer across the duct surface is another mechanism for energy transfer to or from the air inside a duct. It involves conduction through the duct wall and insulation, convection at the inner and outer surfaces, and radiation between the duct and its surroundings. For simplicity, the following discussion excludes the radiation component, which involves complex calculations to evaluate view factors between the ducts and surrounding surfaces. The discussion also assumes that startup transients can be ignored, because HVAC systems in large commercial buildings usually do not cycle on and off during their daily operating periods.

The steady-state heat transfer rate across the duct wall can be determined using heat exchanger effectiveness methods (Stoecker 1980, Gu et al. 1998b):

$$q = \varepsilon \cdot C_{\min} \cdot \left(T_{exterior} - T_{interior}\right) \tag{7}$$

where

- ε: Heat exchanger effectiveness, which is the dimensionless ratio of the actual heat transfer rate to the maximum possible heat transfer rate;
- $C_{\min}$: Heat capacity rate, which is the product of the air mass flow rate inside the duct and the air's specific heat ($c_{p,air}$), W/ºC;
- $T_{exterior}$: Temperature of air surrounding duct exterior, ºC; and
- $T_{interior}$: Temperature of air entering duct, ºC.

Assuming that the heat capacity rate of the air surrounding the duct exterior is infinite (i.e., the temperature of the air surrounding the duct remains approximately constant along the length of the duct), the heat exchanger effectiveness is an exponential relation that depends only on the overall heat transfer coefficient and $C_{\min}$:

$$\varepsilon = 1 - e^{\left(-UA/C_{\min}\right)} \tag{8}$$

The overall duct heat transfer coefficient (neglecting radiation) in Equation 8 can be determined from the sum of the reciprocals of the resistances associated with the conduction and the convection layers:

$$UA_{duct} = \frac{1}{R_{conv,interior}} + \frac{1}{R_{cond}} + \frac{1}{R_{conv,exterior}} \tag{9}$$

Assuming that turbulent forced convection occurs inside the duct, the convection resistance of the internal flow $R_{conv\ interior}$ in Equation 9 can be calculated as:

$$R_{conv,interior} = \frac{1}{h_{conv,interior} \cdot A_{duct}} \tag{10}$$

$R_{conv\ exterior}$ in Equation 9 can be calculated in a similar manner.

An empirical expression for the convection coefficient in Equation 10 is (ASHRAE 2001a):

$$h_{conv,interior} = 0.023 \cdot \frac{k_{air}}{D_h} \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.4} \tag{11}$$

where

A_{duct}: Duct surface area, m^2;

k_{air}: Thermal conductivity of air, W/(m·ºC);

D_h: Duct hydraulic diameter, m;

Re: Reynolds number ($\text{Re} = \rho_{air} \cdot V_{duct} \cdot D_h / \mu_{air}$), dimensionless;

Pr: Prandtl number ($\text{Pr} = \mu_{air} \cdot c_{p,air} / k_{air}$), dimensionless;

ρ_{air}: Air density inside duct, kg/m^3;

V_{duct}: Bulk air velocity through duct, m/s; and

μ_{air}: Air viscosity in duct, $N{\cdot}s/m^2$.

The conduction resistance of the duct wall R_{cond} is dependent on the duct construction material and thickness, and the insulation R-value, and is calculated as the sum of the duct wall resistance and the insulation resistance.

Outside the duct, combined natural and forced convection can occur. Determining a generally applicable combined convection coefficient is difficult because of the wide variation in duct characteristics and environmental conditions that can be found in the large commercial building stock. Based on conditions in residential attics, which are somewhat like ceiling return air plenums, Walker (1993) has suggested that the coefficient can be determined by the following correlation, which makes the larger of the two coefficients most dominant and maintains a smooth transition from one to the other:

$$h_{conv,exterior} = \left(h^3_{natural,exterior} + h^3_{forced,exterior}\right)^{1/3} \tag{12}$$

The forced convection coefficient in Equation 12 can be expressed by the following empirical correlation that has been linearized over the range of temperatures expected in residential attics:

$$h_{forced,exterior} = \left[18.192 - 0.0378 \cdot \left(\frac{T_{duct,surface} + T_{exterior}}{2}\right)\right] \cdot V^{0.8}_{exterior} \tag{13}$$

and the natural convection coefficient can be expressed by another empirical correlation, which uses the same length scale as the forced convection coefficient:

$$h_{natural,exterior} = 3.2 \cdot \left|T_{duct,surface} - T_{exterior}\right|^{1/3} \tag{14}$$

where

$T_{duct,surface}$: Average temperature of duct exterior surface, ºC; and

$V_{exterior}$: Bulk air velocity across duct exterior, m/s.

By definition, the average temperature of the duct exterior surface can be determined as:

$$T_{duct,surface} = T_{exterior} - \left(\frac{q}{h_{conv,exterior} \cdot A_{duct}}\right) \tag{15}$$

In calculating the duct surface heat transfer, an iterative solution is needed to account for the interdependencies between $T_{duct,surface}$, q, and UA_{duct}.

3. REVIEW OF MODEL IMPLEMENTATIONS

3.1 DOE-2.1E

Given that the current Title 24 compliance procedures are based upon DOE-2.1E (Version 110), it useful to describe the duct thermal performance modeling capabilities and limitations of DOE-2.1E. Xu et al. (1999a) have reviewed these capabilities in detail; the following summarizes and expands upon their findings.

The DOE-2 computer simulation program has five major subprograms that are executed in sequence to simulate mass and energy flows in a building:

1. The BDL Processor subprogram translates user input for use in the other four subprograms.
2. The LOADS subprogram calculates the sensible and latent components of the hourly heating or cooling load for each user-designated space in the building.
3. Using the zone loads calculated by the LOADS subprogram, the SYSTEMS subprogram calculates airflow rates, fan power, and coil loads for airside equipment (fans, coils, and ducts).
4. Using the coil loads calculated by the SYSTEMS subprogram, the PLANT section calculates the energy used by primary equipment, such as boilers, chillers, cooling towers, and storage tanks.
5. Based on the energy use calculated in the SYSTEMS and LOADS subprograms, the ECON subprogram calculates the cost of energy.

At a deeper level within the program structure, there are hundreds of FORTRAN subroutines intertwined together like “spaghetti” (Crawley et al. 1998). This non-modular structure has resulted from various program development efforts by many different people over the past three decades. As a result, making even minor changes or improvements to the program is difficult and expensive. This is a significant problem, given that the capabilities of DOE-2.1E to account for duct leakage and surface heat transfer effects are extremely limited.

In DOE-2.1E, up to and including the last official, federal government-sanctioned version of the program (Version 117), the DUCT-AIR-LOSS program keyword is used to account for air leaking out of supply ducts. With this keyword, the user specifies the constant fraction of the system supply airflow that is lost from the ducts, thereby reducing the amount of supply air that reaches the conditioned zones. Surface heat transfer effects for supply ducts are accounted for by the DUCT-DELTA-T keyword, which simply specifies a constant temperature decrease for hot ducts and a constant temperature increase for cold ducts. Energy associated with supply air duct leakage and supply air temperature changes is not included in the building energy balance; instead, it is assumed that energy is transferred directly between the supply ducts and outdoors, regardless of where the ducts are actually located (Buhl et al. 1981). Effectively, there is no duct model within the reference version of DOE-2.1E; the user must use an external model or engineering judgment to determine values for these keywords.

According to Xu et al. (1999a), later proprietary versions of DOE-2.1E were improved so that the user could specify a supply duct heat loss coefficient instead of simply a supply duct temperature change. The building space that “receives” the supply duct leakage air and that is linked to the surface heat transfer effects could also be specified using a new keyword: PIPE&DUCT-ZONE. Typically, this is a ceiling return plenum (specified by the keyword ZONE-TYPE = PLENUM), but could be an unconditioned space. If the plenum or unconditioned space is adjacent to conditioned zones, the zone loads must be determined by specifying SIZING-OPTION = ADJUST-LOADS for the conditioned spaces under the ZONE command. In doing so, the adjacent zones will have their load calculations adjusted for changes in the plenum or unconditioned space temperature.

With the newer versions of DOE-2.1E, if the supply and return flows are both ducted, a different modeling strategy is required than the one outlined above. In this case, the space surrounding the ducts should *not* be modeled as a ZONE-TYPE = PLENUM, but should be modeled as a ZONE-TYPE = UNCONDITIONED instead. This allows the user to specify the RETURN-AIR-PATH keyword to be DUCT without its value being overwritten (which occurs when ZONE-TYPE = PLENUM).

In spite of the improvements made recently, significant limitations for modeling duct thermal performance in large commercial buildings remain in DOE-2.1E:

- The supply duct air leakage rate is modeled as a fixed fraction for all system flow rates and types. While this assumption is plausible for some CAV systems, it is not consistent with the conditions that occur in duct sections upstream of terminal boxes in a VAV system. Furthermore, leakage cannot currently be divided into upstream and downstream components relative to the location of terminal boxes, which means the effects of these different types of leakage on terminal box reheat loads and fan power cannot be properly modeled.
- The supply duct heat loss coefficient is constant and does not include radiation effects, or the dependence of convection on airflows and duct interior-exterior air temperature differences. The impact of these assumptions for ducts in large commercial buildings is not clear. However, for ductwork located in an unconditioned rooftop ceiling space, the effect of radiation might be significant (Hirsch 1996). Also, in the duct heat transfer calculation, heat exchanger analysis methods are not used to account for the variation in air temperature along the length of duct (which occurs in real systems); instead, the heat transfer rate is simply based on the average supply-air temperature.
- Duct leakage and surface heat transfer effects are not modeled for return ducts.

3.2 Other Commercially Available Simulation Programs

The U.S. Department of Energy (DOE 2003a) lists 256 software tools for energy-related analyses of buildings. Only a small number of these programs can simulate the whole-building energy use of HVAC systems in large commercial buildings, on an hourly or sub-hourly time scale; even fewer have open source code that is publicly available for review and modification. Other than the two DOE-2.1E Version 110 based programs (EnergyPro and Perform95) identified earlier in this report, none are certified for use in Title 24 compliance analyses.

The following describes three notable programs that can be used now for energy analyses of large commercial buildings, and that have duct thermal performance models beyond the capabilities of DOE-2.1E or that might be adapted to include such models because of their expanded features and other modeling capabilities.

3.2.1 DOE-2.2

In 1993, the Simulation Research Group of Lawrence Berkeley National Laboratory (LBNL) and James J. Hirsch & Associates (JJH) started developing a new version of the DOE-2 building energy simulation program. This new version, to be called DOE-2.2, was intended to replace the current version, DOE-2.1E. Funding the development efforts were the U.S. Department of

Energy (DOE), the Electric Power Research Institute, and others. For various reasons, DOE and LBNL are no longer involved in developing DOE-2.2; however, JJH has continued to develop the program and a proprietary beta version is now available for testing.

Two significant improvements to the duct leakage and surface heat transfer algorithms in the latest versions of DOE-2.1E have been made in DOE-2.2. These include:

- Air leakage from return ducts is now modeled, although surface heat transfer for return ducts is still not modeled.
- The supply duct surface heat transfer algorithm is now based on a heat exchanger model.

Even with these improvements, most of the significant limitations for modeling duct thermal performance in large commercial buildings that were identified in DOE-2.1E remain in DOE 2.2.

3.2.2 EnergyPlus

Since 1996, DOE has been funding LBNL, the University of Illinois, the U.S. Army Construction Engineering Research Laboratory, and others to develop EnergyPlus. This new program is intended to replace both DOE-2 and BLAST (a whole-building energy analysis program sponsored by the Department of Defense). It builds on the strengths of DOE-2 and BLAST by combining the best attributes of both hour-by-hour energy analysis programs into a modular program that can be more easily maintained and upgraded. EnergyPlus includes a number of innovative simulation features (DOE 2003b, 2003c), such as variable time steps (as small as 1 minute) and modular system and plant models that are solved simultaneously with a heat balance-based loads simulation. Within the PIER High Performance Commercial Building Systems program, UC San Diego is completing work on a two-node-zone displacement ventilation model for EnergyPlus to deal with the effects of room air mixing, jets, and buoyancy on space temperature distribution and thermal comfort. All of these features make EnergyPlus attractive for evaluating innovative low-energy cooling building designs.

EnergyPlus has no duct thermal performance models to account for duct leakage or surface heat transfer effects. However, the modular nature of EnergyPlus and a planned link with the SPARK equation-based component simulation tool (Crawley et al. 2001, LBNL 2002) should make model integration relatively straightforward once the models have been defined. The DOE "Guide for Module Developers: Everything You Need to Know about EnergyPlus Calculational Development" (DOE 2003e) provides explicit instructions on how to proceed with module development and integration. The Florida Solar Energy Center is currently integrating a residential duct thermal performance model (Gu et al. 1998b) into EnergyPlus, but it is unlikely that this model will be able to deal with the complex duct systems and operation strategies in large commercial buildings without further development.

A link with COMIS (a multizone airflow network simulation program) is included in EnergyPlus to determine time-varying envelope and interzonal flows. However, at this time, duct flows are either user-specified or are determined solely based on thermal requirements, rather than being determined using COMIS flow and pressure correlations. COMIS (like other similar programs) cannot accurately model duct components involving multi-port flow junctions, because it assumes that duct flows depend on zonal pressure differences, but not directly on flows in other branches (Lorenzetti 2001). In real systems, duct flows (and pressure loss coefficients) in adjacent duct branches can be interdependent.

Assuming that a reliable airflow network solution could be developed to model duct flows in EnergyPlus, there is still the question of appropriate inputs and who would use such models. The location and size distribution of leaks in a duct system are practically unknowable, which means there may be little point to conducting detailed duct airflow-pressure simulations in other than research investigations. Even if detailed leakage data could be gathered in a real building (likely at great expense), these data would be so voluminous that the burden of entering them might dissuade users from using the model. For compliance evaluations involving hypothetical buildings, large amounts of input data are undesirable for the same reason. Simplifying assumptions could be made about leak location and size distribution (e.g., fixed leakage flows, or fixed fractional leakage flows, as appropriate to various duct sections). Bayesian data analysis methods could also be used to identify key input data (and to develop measurement and performance verification protocols); however, these techniques are only now being developed for use in whole-building airflow simulations (Sohn et al. 2000), and the size and distribution of duct leaks in the large commercial building stock is not yet well defined.

3.3.3 TRNSYS

The TRNSYS simulation program is a transient simulation program that has been in use since 1975, primarily to model building thermal energy systems in research applications (Klein et al. 1996). Because of its modular nature, it allows substantial flexibility for a user to specify a building and its HVAC system component by component. User-specified parameters describe the characteristics of each component, and user-specified interconnections between inputs and outputs link the components. Simulations involve the simultaneous solution of large systems of equations contained in the FORTRAN subroutines that describe the specified component models. Each subroutine defines a component model; user inputs describe to TRNSYS which subroutines should be linked and executed to define the thermal system of interest. This simulation strategy is distinctly different from the sequential solution, predefined system schematic strategy used in programs such as DOE-2.

Many component models are already available in TRNSYS for simulating HVAC system performance in large commercial buildings, and are well documented using the same source code style that is used in the ASHRAE HVAC Secondary Toolkit (Brandemuehl et al. 1993). Franconi (1999) used TRNSYS to model the effects of duct leakage on VAV system performance for a 10-story office building. Problem specific equipment and control models were developed using FORTRAN subroutines from the ASHRAE Toolkit (e.g., a VAV fan, an air-side economizer). New models were also created to represent duct air leakage, VAV boxes with reheat coils, and ceiling return plenums.

The Franconi simulation approach involved a quasi-steady-state strategy with a one-hour time step, and can be described by three sequential steps:

1. Hourly zone loads (heat extraction and addition rates) and zone air temperatures are calculated using DOE-2, for a constant air volume (CAV) system that has no duct leakage. These results are then output to a data file, which is read as input by TRNSYS. The data file also includes the corresponding hourly weather conditions, latent heat gains in conditioned spaces, and heat input to the ceiling plenum from lights.
2. TRNSYS generates hourly HVAC system fan and coil energy consumption data using interconnected detailed component models for the heating and cooling coils, fans, ducts, terminal boxes, economizer, and return plenum. The solution for each hour involves numerous iterations that terminate when convergence is achieved; convergence occurs when the error tolerances associated with component input and output variables are satisfied. Various duct leakage configurations are modeled at this stage. The TRNSYS analysis considers only hours when the HVAC system is operating.

3. Regression analyses based on correlations between DOE-2 system and plant energy use are used to translate the TRNSYS system level coil load data to plant level energy use; energy costs are subsequently calculated based on this energy use.

In this approach, the distribution system simulation is uncoupled from the loads and plant simulations of DOE-2, in the same manner that DOE-2 itself uses. The difference is that the TRNSYS system simulation expands beyond DOE-2 modeling capabilities to offer more flexibility in modeling duct thermal performance issues. A more rigorous approach would involve a coupled simultaneous solution of the loads, system, and plant performance, which could be done now using EnergyPlus if the TRNSYS models were integrated with that program.

Appendix II contains the FORTRAN source code implementations of the seven models that Franconi used in TRNSYS (TYPES: 70, Fan (supply and return); 75, Cooling Coil; 77, Zone Return Air Mixing; 80, Economizer; 81, Ceiling Return Plenum; 82, Upstream Ducts; and 86, VAV Box and Downstream Ducts). For each model, a subroutine and function call mapping is provided to help the reader follow the program logic. Then, the source code begins by defining the purpose of the model, the input and output variables, and the parameters used to characterize component performance. Finally, the source code that follows defines component performance and is essentially self-documenting.

For the most part, the seven models that Franconi used reflect the modeling principles discussed earlier in this report. The exception is that duct surface heat transfer effects are not fully or properly addressed; preliminary efforts to model these effects were included in the subroutine for ducts upstream of VAV boxes (TYPE82), but were never tested or used. Also, it appears that some of the assumptions and duct surface heat transfer equations defined in that subroutine are incorrect (particularly when ducts are insulated) and require further development. Furthermore, the ceiling return plenum model (TYPE81) does not include a storage term to account for the thermal mass of the concrete ceiling of the plenum. Excluding the storage term means that the amplitude of return air temperature variations in the plenum might be larger than actually occurs, especially when the air-handling system is off, and may also be improperly phased as well. The impacts of this omission on duct surface heat transfer rates during system operation are not clear and require further investigation. However, it likely has no significant impact on the most important parameter affected by supply duct air leakage: fan power consumption.

Figure 1 shows a sample of the performance parameters calculated by TRNSYS using the seven models for one hour of VAV system operation during the cooling season, for a case with 10% duct leakage upstream and 10% duct leakage downstream of the VAV boxes (leakage paths shown by dashed lines leading to ceiling plenum). This graphic representation of the TRNSYS output is based on a spreadsheet recently developed by the author as an aid to understand and test the model, and is not generated by TRNSYS itself.

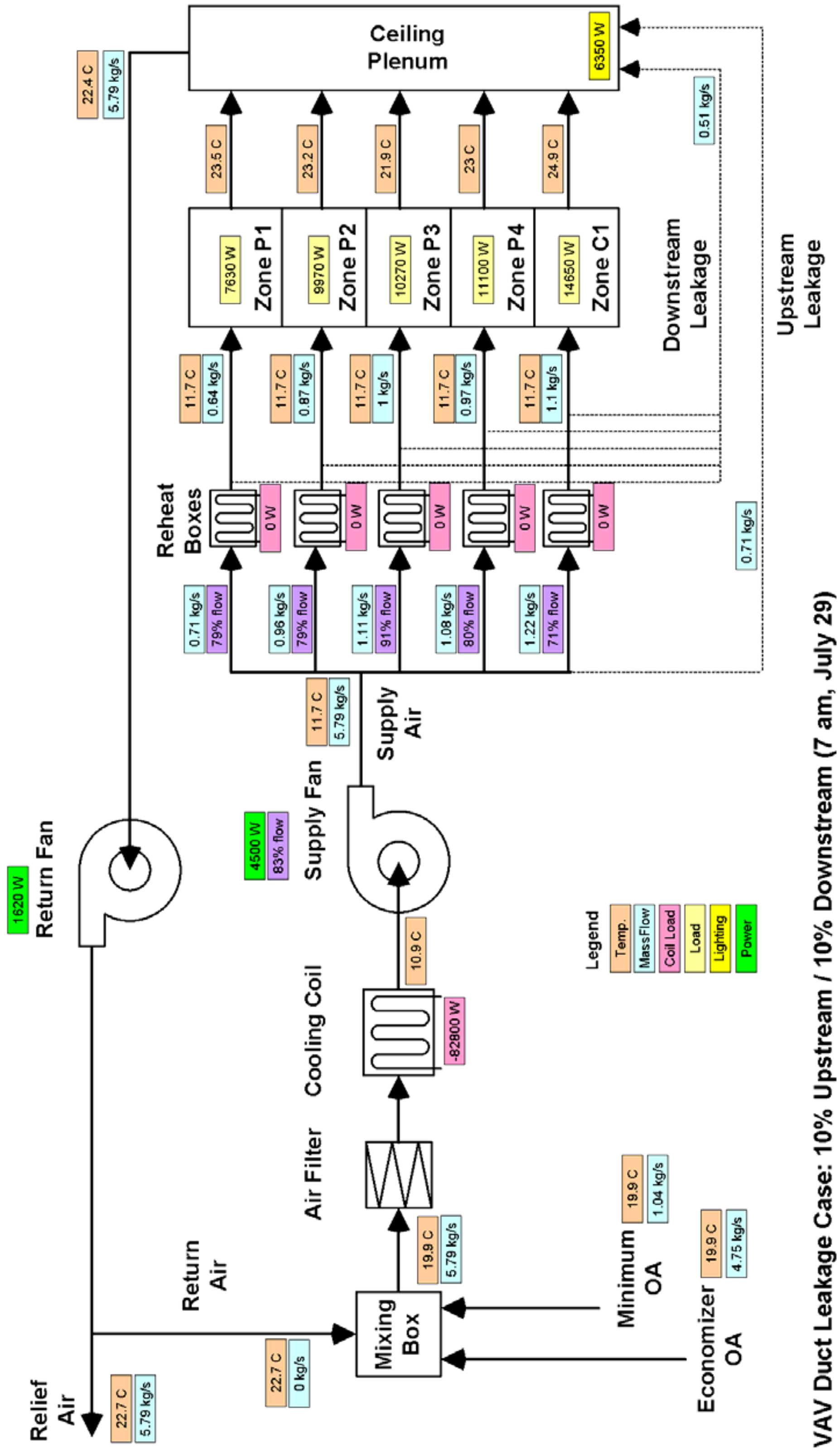

Figure 1. Sample TRNSYS Output

4. PROJECT OUTCOMES

Our duct modeling review involved a literature review of 188 documents related to past HVAC system modeling efforts. We supplemented the review through discussions with building simulation experts to assess new, unpublished, relevant work. Based on this review, we have the following two recommendations for duct modeling approaches: one is for a short-term approach that can be used in the impacts analysis task of this project; the other is for the long-term development of an improved compliance tool for Title 24.

4.1 Recommended Short-Term Modeling Approach

We conclude that the best approach for our impacts analysis task in this project is to build upon the Franconi (1999) research that used DOE-2 and TRNSYS to evaluate HVAC system performance. An advantage of using the DOE-2/TRNSYS approach in this project is that DOE-2 prototypical models for a large commercial California building are already available, as are the custom TRNSYS component models. Another advantage is that the duct leakage modeling approach and its results for a California building have already been validated, and no substantial changes to the simulation tool are required to carry out our analyses. No other whole-building modeling approach to assess duct system performance for large commercial buildings is currently as advanced as this approach.

4.2 Recommended Long-Term Modeling Approach

The choices for incorporating the impacts of duct performance in large commercial buildings into the Title-24, Non-Residential Building Energy Standards include:

1. Using the current DOE-2.1E program,
2. Using an add-on calculation routine along with DOE-2, or
3. Using an alternative calculation method or simulation program.

Initially, it would seem that using the first path might be the most appropriate. The DOE-2.1E program is well entrenched into the Title-24 compliance path, and most importantly, it is used to benchmark alternative compliance models, which means that unless the DOE-2.1E program gets the correct answer, alternative programs that do get the right answer will not be certified. However, as described in this report, there are a number of modeling problems in DOE-2.1E that need to be addressed. Because DOE no longer supports the development of DOE-2, it is likely that modifications would need to be made in the private sector, which could result in proprietary source code that might not be available for public inspection. Furthermore, the convoluted structure of DOE-2 will make modifications difficult and expensive, which is one of the reasons DOE decided to pursue the development of EnergyPlus.

The second path provides a possible alternative. However, the sequential DOE-2 / TRNSYS modeling approach could best be described as “user hostile”. Even though an Excel spreadsheet “interface” has been recently developed to help organize and visualize the input and output of TRNSYS, this simulation approach remains inappropriate outside a research environment. It is unlikely this approach would be practical on a day-to-day basis for compliance analyses.

Assuming that EnergyPlus could be certified as a compliance tool for use in support of the 2008 revisions to Title 24, we suggest that the long-term strategy should involve integrating duct thermal performance models with EnergyPlus. This long-term approach focuses on EnergyPlus rather than on the current compliance version of DOE-2, because we expect that the recommended enhancements could be more easily applied and used in EnergyPlus for future analyses of innovative low-energy cooling designs. In particular, although EnergyPlus at this time has no capabilities to model duct system thermal losses, we expect that the TRNSYS HVAC system models or ones like them could be incorporated into EnergyPlus directly or through a planned SPARK link to provide a more practical integrated tool for designers.

During the development of EnergyPlus, a link to TRNSYS was planned; such a link would make it easier to add the current TRNSYS models to EnergyPlus. However, it is unlikely that this link will be established, because TRNSYS and EnergyPlus each have separate means to determine simulation time steps, and a way to link these time steps in an external-coupling run-time mode has not been resolved. Essentially, one of the two programs needs to be the "brain" running the simulation and TRNSYS is not setup to give control to EnergyPlus for this functionality. As a result, simply linking the TRNSYS duct thermal performance models to EnergyPlus is not a practical option at this time.

Apart from needing to be certified as an alternative calculation method, a key issue related to using EnergyPlus as a compliance tool is whether users could easily operate the program. At this time, a rudimentary user-interface has been developed for EnergyPlus, but is not yet publicly available. If EnergyPlus is to be used in support of the 2008 Standards, then the development of this interface needs to be accelerated. A related issue is that there is already a well-established user base that knows how to operate DOE-2.1E for compliance analyses. To make it easy for current DOE-2 (and BLAST) users to move to EnergyPlus, utilities are already being developed to convert BLAST and DOE-2 input files for use in EnergyPlus.

4.3 Recommendations for Future Work

Incorporating duct-modeling capabilities into compliance tools is only one aspect of the changes that need to be made to the non-residential standards. Before duct performance in large commercial buildings can be accounted for in Title 24 nonresidential building energy standards, there are several other issues that must be addressed and resolved. These include:

1. Specifying reliable duct air leakage measurement techniques that can be practically applied in the large commercial building sector.
2. Defining the duct leakage condition for the standard building used in Title 24 compliance simulations.
3. Assuring consistency between simulated duct performance impacts and actual impacts.
4. Developing compliance tests for the Alternative Calculation Method (ACM) Approval Manual (CEC 2001b) to evaluate duct performance simulations.

Regarding Issues 1 and 2, new duct air leakage measurement techniques for large commercial buildings are already under development at LBNL. These efforts are focused on developing a rapid technique that measures leakage flows rather than leakage area, and we expect that it could be used to populate a database of duct leakage conditions in the existing building stock.

After the "typical" duct leakage for the building stock is defined, then a decision can be made about what duct leakage level to assign to the standard building. If the standard building description includes a typical duct air leakage rate, then proposed buildings will be rewarded for sealing ducts. If instead the standard building has a reduced leakage level, proposed buildings that are not sealed will be penalized. The decision about what leakage level to assume for the standard building description will depend upon the preparedness of the market to handle required duct efficiency improvements, as opposed to optional improvements.

In terms of prescriptive compliance options, if the standard-building duct performance parameters are established to correspond to typical duct air leakage, determining compliance using the prescriptive approach is straightforward. If the proposed building has a typical duct air leakage level and has ducts insulated to Title 24 requirements, the building complies with respect to ducts. In other words with nothing done to improve duct performance in the building, it would meet the minimal duct performance level in this case. On the other hand, if the standard building has tighter-than-typical duct air leakage specifications, then compliance would require either performance measurements (i.e., duct air leakage measurements), or increased energy efficiency of other building components.

With the standard building defined as having leaky ducts, improving the duct performance in the proposed building affects compliance only if the performance budget approach is used. If leaks are sealed as a compliance conservation measure, standardized testing methods must be adopted to verify reduced leakage rates. Leakage rates determined from the tests would be part of the duct performance input data in the performance compliance analysis for the proposed building.

For Issue 3, one study has already shown through detailed minute-by-minute field measurements in a large commercial building that duct leakage has a significant impact on HVAC system performance (Diamond et al. 2003). The extensive set of HVAC system performance data collected by Diamond et al. could be used to validate simulation tools that are used to predict the duct performance impacts.

Regarding Issue 4, several tests must be performed already on alternative calculation methods before they are approved. Although a test does not yet exist, the proper modeling of duct performance in these alternative methods should be evaluated as part of these capability tests. Given that the current two certified nonresidential compliance tools depend upon DOE-2.1E as the reference evaluation program, and that DOE-2.1E cannot properly account for duct thermal performance, it is expected that results obtained using an alternative calculation method that properly accounts for duct thermal performance might differ substantially from the reference program results. Thus, we recommend that a new reference program be identified for use at least in this test (e.g., EnergyPlus). A prerequisite in this case is that the reference method be appropriately validated against field measurements.

When this project is complete, we expect that it will successfully demonstrate to the building industry that duct leakage in commercial buildings is an important performance issue, and that there is value in reducing thermal losses associated with this leakage. It will also provide the basis for the development of standards that address thermal deficiencies in large commercial duct systems. Three additional steps will be required to further develop duct-modeling capabilities that address limitations in existing models and to initiate strong market activity related to duct system improvements. We recommend that these steps include:

1. Implementing duct models in user-friendly commercially-available software for building energy simulation, validating the implementations with case studies and demonstrations, and obtaining certification for software use as a primary or alternative compliance tool in support of the Title 24 Nonresidential Standards.
2. Developing methodologies to deal with airflows entering VAV boxes from ceiling return plenums (e.g., to model parallel fan-powered VAV boxes), to deal with duct surface heat transfer effects, and to deal with static pressure reset and supply air temperature reset strategies.
3. Transferring information to practitioners through publications, conferences, workshops, and other education programs.

GLOSSARY

ACM	Alternative Calculation Method
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CAV	Constant Air Volume
CEC	California Energy Commission
DOE	U.S. Department of Energy
EIA	Energy Information Administration
GWh	Giga Watt hours, 10^9 Wh, 10^6 kWh
HVAC	Heating, Ventilating, and Air Conditioning
JJH	James J. Hirsch & Associates
LBNL	Lawrence Berkeley National Laboratory
MW	Mega Watt, 10^6 W
PIER	Public Interest Energy Research
UC	University of California
VAV	Variable Air Volume

APPENDIX I: BIBLIOGRAPHY

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APPENDIX II: TRNSYS DUCT PERFORMANCE SUBROUTINES

Subroutine TYPE 70: Fan (Supply or Return)

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SUBROUTINE AND FUNCTION CALL MAPPING
Calculate the fan power and leaving air temperature and humidity for fan using simple
part load characteristics.
        SUBROUTINE TYPE70 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK -- subroutine in TRNWIN\Kernal\rcheck.for

           hEnt = ENTHALPY5(CPAIR,HFG,CPVAP,TEnt,WEnt) -- F2 in type 70
           rho = RHODRY(PATM,RAIR,TABSADD,TEnt,WEnt) -- F3 in type 70
           TLvg = DRYBULB5(CPAIR,CPVAP,HFG,hLvg,WLvg) -- F1 in type 70

        F1 REAL FUNCTION DRYBULB5 (CPAIR,CPVAP,HFG,H,W)
             Calculate the dry bulb temperature of moist air from enthalpy
             and humidity.

        F2 REAL FUNCTION ENTHALPY5 (CPAIR,HFG,CPVAP,TDB,W)
             Calculate the enthalpy of moist air.

        F3 REAL FUNCTION RHODRY (PATM,RAIR,TABSADD,TDB,W)
             Calculate dry air density.

      PATM     =  101325.0       Atmospheric pressure (Pa)
      CPAIR    =    1006.0       Specific heat of dry air (J/kg C)
      CPVAP    =    1805.0       Specific heat of saturated water vapor (J/kg C)
      HFG      = 2501000.0       Latent heat of vaporization of water (J/kg)
      RAIR     =     287.055     Gas constant for air (J/kg C)
      TABSADD  =     273.15      Additive factor to convert user P to Kelvin:
                                 tKel = Prop(TKelMult)*T + Prop(TKelAdd)


SOURCE CODE
      SUBROUTINE TYPE70 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C     SUBROUTINE FANSIM (Prop,P,M,TEnt,WEnt,
C     &                   TLvg,WLvg,Power,ErrStat)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*     UPDATED FOR THE TRNSYS-LIBRARY BY RUEDIGER SCHWARZ AND NATE BLAIR
C*******************************************************************
C*    SUBROUTINE: FANSIM
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the fan power and leaving
C*                air temperature and humidity for fan
C*                using simple part load characteristics.
C*******************************************************************
C*    INPUT VARIABLES DESCRIPTION(UNITS)                    SAMPLE VALUE
C*    XIN(1)    M     Dry air mass flow rate(kg/s)             3.4
C*    XIN(2)  TEnt    Entering air dry bulb temperature(C)    12.78
C*    XIN(3)  WEnt    Entering air humidity ratio(-)            .00835
C*
C*    OUTPUT VARIABLES
C*    OUT(1)  TLvg    Leaving air dry bulb temperature(C)     14.3358
C*    OUT(2)  WLvg    Leaving air humidity ratio(-)             .00835
C*    OUT(3)  Power   Fan power(W)                          5401.19
C*    OUT(4) ErrStat  Error status indicator,0=ok,1=error(-)   0.0
C*    OUT(5)   PLR    Part load ratio (-)
C*
C*    Note:   If M<0, TEnt and WEnt are assumed to be fan outlet air
C*            conditions, TLvg and WLvg are calculated inlet conditions
```

```
C*
C*    PARAMETERS
C*    PAR(1) EffMot    Motor drive efficiency(-)                       .85
C*    PAR(2) MotorLoss Fraction of motor heat loss to fluid stream  1.0
C*    PAR(3) FlowRated Rated volumetric flow rate(m3/s)              5.664
C*    PAR(4) PowRated  Rated shaft power(W)                      17700.0
C*    PAR(5) PlrContl  Mode for fan control(-)                       3.0
C*                     PlrContl = 1:  Discharge dampers
C*                     PlrContl = 2:  Inlet vanes
C*                     PlrContl = 3:  Variable speed drive
C*                     PlrContl = 4:  Variable speed drive w/ constant s.p. control
C*                     PlrContl = 5:  Cubic
C*********************************************************************
C     MAJOR RESTRICTIONS:     Fan curve fits are independent of pressure
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          fanpmp.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       DRYBULB5
C                             ENTHALPY5
C                             RHODRY
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              BLAST.  1986.  Building Loads Analysis
C                             and System Thermodynamics Program:
C                             User's Manual, Version 3.0. U.S. Army
C                             Construction Engineering Research
C                             Laboratory, Champaign, IL. pp.5-26-5-27.
C*********************************************************************
C     INTERNAL VARIABLES:
C     effFan        Fan efficiency                                 (-)
C     hEnt          Entering air enthalpy                       (J/kg)
C     rho           Entering moist air density                 (kg/m3)
C     fflp          Fraction of full-load fan power                (-)
C     plr           Part load flow ratio                           (-)
C     powShaft      Shaft power                                    (W)
C     qLoss         Heat transfer to fluid stream                  (W)
C     c(i,PlrContl) Regression coefficients
C     small         Small number used in place of zero
C*********************************************************************

C    $INCLUDE: 'fanpmp.inc'
      DOUBLE PRECISION XIN, OUT

       DIMENSION XIN(3), OUT(5), PAR(5)
      DIMENSION C(4,5), INFO(15)

       INTEGER ErrStat, IOPT, NI, NP, ND, INFO

       REAL M, PAR

       CHARACTER*3 YCHECK(3), OCHECK(5)

       DATA YCHECK/'MF2','TE1','DM1'/
      DATA OCHECK/'TE1','DM1','PW2','DM1','DM1'/
      DATA PATM/101325.0/,CPAIR/1006.0/,CPVAP/1805.0/,HFG/2501000/,
     &     RAIR/287.055/, TABSADD/273.15/

      IOPT  = -1.
      NI    = 3.      !CORRECT NUMBER OF INPUTS
```

```
      NP    = 5.      !CORRECT NUMBER OF PARAMETERS
      ND    = 0.      !CORRECT NUMBER OF DERIVATIVES

      M       = XIN(1)
      TENT    = XIN(2)
      WENT    = XIN(3)

      TLVG    = OUT(1)
      WLVG    = OUT(2)
      POWER   = OUT(3)
      ERRSTAT = OUT(4)
      plr     = OUT(5)

      EFFMOT    = PAR(1)
      MOTORLOSS = PAR(2)
      FLOWRATED = PAR(3)
      POWRATED  = PAR(4)
      PLRCONTL  = PAR(5)

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
C      CHECKS #S IN USER SUPLLIED INFO ARRAY W/ NI, NP, AND ND
        CALL RCHECK(INFO,YCHECK,OCHECK)
C     CHECKS TO SEE IF THE UNITS ARE CONSISTENT
        INFO(6)=4
      ENDIF

C      DIMENSION P(NPFANPMP)

C2*** Set regression coefficients for fraction of full load power
C2***        Discharge dampers
      DATA C/0.3507123,   0.3085, -0.54137, 0.871988,
C2***        Inlet vanes
     &          0.3707,   0.9725,  -0.3424,      0.0,
C2***        Variable speed
     &         0.00153, 0.005208,   1.1086, -0.11635563,
C2***        Variable speed w/ constant static pressure control
     &         0.00441, 0.28808,    0.2626,  0.4498,
C2***        Cubic
     &             0.0,     0.0,       0.0,     1.0/
      DATA small/1.E-9/

      ErrStat = 0

C1*** If flowrate is zero, power is zero
      IF(M .EQ. 0) THEN
        TLVG=TENT
        WLVG=WENT
        POWER=0
        plr=0
        GOTO 999
      ENDIF

C1*** Calculate entering moist air properties
      hEnt = ENTHALPY5(CPAIR,HFG,CPVAP,TEnt,WEnt)
      rho = RHODRY(PATM,RAIR,TABSADD,TEnt,WEnt)

C1*** Calculate the part load ratio based on rated flow
      plr=ABS(M)/rho/FlowRated

C1*** Calculate the fraction of full-load power based on rating point
C2*** fflp = c(1) + c(2)*plr + c(3)*plr**2 + c(4)*plr**3
C2*** Regression coefficients, c(i), vary with control mode
      fflp =c(1,PlrContl)+plr*( c(2,PlrContl)
     &                   +plr*( c(3,PlrContl)
     &                   +plr*  c(4,PlrContl) ) )
```

```
C1*** Calculate the actual fan shaft power and motor power
      powShaft = PowRated*fflp
      Power = powShaft/EffMot

C1*** Calculate the leaving air conditions
C2*** If flow is zero, ABS(M) < small, the value of M is replaces with
C2    small of the same sign as M in calculating hLvg
      qLoss = powShaft + (Power-powShaft)*MotorLoss
      hLvg = hEnt + qLoss/SIGN(MAX(ABS(M),small),M)
      WLvg = WEnt
      TLvg = DRYBULB5(CPAIR,CPVAP,HFG,hLvg,WLvg)

  999 Continue

      OUT(1) = TLVG
      OUT(2) = WLVG
      OUT(3) = POWER
      OUT(4) = ERRSTAT
      OUT(5) = plr

      RETURN 1
      END

      REAL FUNCTION DRYBULB5 (CPAIR,CPVAP,HFG,H,W)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: DRYBULB5
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the dry bulb temperature of
C*              moist air from enthalpy and humidity.
C***********************************************************************
C*    INPUT VARIABLES:
C*    H             Enthalpy                                      (J/kg)
C*    W             Humidity ratio                                   (-)
C*
C*    OUTPUT VARIABLES:
C*    Drybulb5       Dry bulb temperature                             (C)
C*
C*    PROPERTIES:
C*    CpAir         Specific heat of air                        (J/kg C)
C*    CpVap         Specific heat of water vapor                (J/kg C)
C*    Hfg           Reference heat of vaporization of water       (J/kg)
C***********************************************************************
C     MAJOR RESTRICTIONS:     Uses perfect gas relationships
C                             Fit for enthalpy of saturated water vapor
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          PROP.INC
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************

C $INCLUDE: 'prop.inc'
```

```
C1*** Calculate the dry bulb temperature as a function of enthalpy and
C1*** humidity ratio.
C2*** hDryAir = Prop(CpAir)*TDB
C2*** hSatVap = Prop(Hfg) + Prop(CpVap)*TDB
C2*** Enthalpy = hDryAir + W*hSatVap

      Drybulb5 = (H-Hfg*W)/(CpAir+CpVap*W)

      RETURN
      END

      REAL FUNCTION ENTHALPY5 (CPAIR,HFG,CPVAP,TDB,W)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: ENTHALPY5
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the enthalpy of moist air.
C***********************************************************************
C*    INPUT VARIABLES:
C*    TDB           Dry bulb temperature                            (C)
C*    W             Humidity ratio                                  (-)
C*
C*    OUTPUT VARIABLES:
C*    Enthalpy      Enthalpy of moist air                        (J/kg)
C*
C*    PROPERTIES:
C*    CpAir         Specific heat of air                       (J/kg C)
C*    CpVap         Specific heat of water vapor               (J/kg C)
C*    Hfg           Reference heat of vaporization of water      (J/kg)
C***********************************************************************
C     MAJOR RESTRICTIONS      Uses perfect gas relationships
C                             Fit for enthalpy of saturated water vapor
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          PROP.INC
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************

C $INCLUDE: 'prop.inc'

C1*** Calculate the enthalpy as a function of dry bulb temperature and
C1*** humidity ratio.

      hDryAir = CpAir*TDB
      hSatVap = Hfg + CpVap*TDB
      Enthalpy5 = hDryAir + W*hSatVap

      RETURN
      END

      REAL FUNCTION RHODRY (PATM,RAIR,TABSADD,TDB,W)
C***********************************************************************
```

```
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*    FUNCTION: RHODRY
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate dry air density.
C*******************************************************************
C*    INPUT VARIABLES
C*    TDB           Dry bulb temperature                           (C)
C*    W             Humidity ratio                                 (-)
C*
C*    OUTPUT VARIABLES
C*    RhoDry        Density of dry air                         (kg/m3)
C*
C*    PROPERTIES
C*    Patm          Atmospheric pressure                          (Pa)
C*    RAir          Gas constant for air                      (J/kg C)
C*    TAbsAdd       Additive constant to convert user T to absolute T
C*******************************************************************
C     MAJOR RESTRICTIONS:     Perfect gas relationships
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*******************************************************************
C     INTERNAL VARIABLES:
C     pAir          Partial pressure of dry air                   (Pa)
C*******************************************************************

C $INCLUDE: 'prop.inc'

C1*** Calculate the dry air density from perfect gas laws.

      pAir = 0.62198*Patm/(0.62198+W)
      RhoDry = pAir/RAir/(TDB+TAbsAdd)

      RETURN
      END
```

Subroutine TYPE 75: Cooling Coil

```
SUBROUTINE AND FUNCTION CALL MAPPING
Model the performance of a counterflow crossflow cooling coil. The model accounts for
condensation on the outside surface. Three conditions are possible:  all wet,
partially wet or all dry. Input includes outlet air setpoint temperature. Water flow
rate is changed until desired value is achieved. Output includes outlet air
temperature and humidity, outlet water temperature, sensible and total cooling
capacities and the wet fraction of air-side surface area.
        SUBROUTINE TYPE75 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for

           hAirRat = ENTHALPY3(Prop,TAirRat,WAirRat) -- function F2 in type 75
           hLiqRatSat = ENTHSAT(Prop,TLiqRat) -- function F4 in type 75
           wAirLvgRat = HUMTH(Prop,TAirRat,hDummy) -- function F6 in type 75
           tAirLvgRat = DRYBULB3(Prop,hAirLvgRat,wAirLvgRat) -- function F13 in type
75
           tDewRat = DEWPOINT(Prop,WAirRat) -- function F3 in type 75
           cpSat = (ENTHSAT(Prop,tDewRat)-hLiqRatSat) -- function F4 in type 75
     &        /(tDewRat-TLiqRat)
           uaH = UAHX(capAirH,hAirRat,capLiqH,hLiqRatSat,QTotRat, -- function F1
                                                                    in type 75
     &        configHX,ErrStat)
           CALL BYPASS(Prop,TairRat,WAirRat,tAirLvgRat,wAirLvgRat, -- subroutine S5
                                                                     in type 75
     &        tAdpRat,wAdpRat,bfRat,ErrStat)
           hAdpRat = ENTHALPY3(Prop,tAdpRat,wAdpRat) -- function F2 in type 75
           tDewPt = DEWPOINT (Prop,WAirEnt) -- function F3 in type 75
           CALL DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt, -- subroutine S1
                                                                   in type 75
     &        UATot,configHX,
     &        TLiqLvg,TAirLvg,WAirLvg,QTot,ErrStat)
           CALL WETCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt, -- subroutine S2
                                                                   in type 75
     &        UAInt,UAExt,configHX,
     &        TLiqLvg,TAirLvg,WAirLvg,QTot,QSen,FWet,
     &        tSurfEnt,ErrStat)
           mLiq = XITERATE(mLiq,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                              in type 75
           hAirEnt = MAIR * ENTHALPY3(Prop,TAirEnt,WAirEnt) -- function F2 in type 75

        S1 SUBROUTINE DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                 UA,ConfigHX,
     &                 TLiqLvg,TAirLvg,WAirLvg,Q,
     &                 ErrStat)
              Calculate the performance of a sensible air-liquid heat exchanger.
              Calculated results include outlet air temperature and humidity, outlet
              water temperature, and heat transfer rate.
           CALL HEATEX (capLiq,TLiqEnt,capAir,TAirEnt,UA,ConfigHX, -- subroutine S3
                                                                     in type 75
     &        TLiqLvg,TAirLvg)

        S2 SUBROUTINE WETCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                 UAIntTot,UAExtTot,ConfigHX,
     &                 TLiqLvg,TAirLvg,WAirLvg,QTot,QSen,FWet,
     &                 TSurfEnt,ErrStat)
              Calculate the performance of a cooling coil when the external fin
              surface is complete wet. Results include outlet air temperature and
              humidity, outlet water temperature, sensible and total cooling
              capacities, and the wet fraction of the air-side surface area.

           hAirEnt = ENTHALPY3(Prop,TAirEnt,WAirEnt) -- function F2 in type 75
           hLiqEntSat = ENTHSAT(Prop,TLiqEnt) -- function F4 in type 75
           tDewEnt = DEWPOINT(Prop,WAirEnt) -- function F3 in type 75
```

```
          cpSat = (ENTHSAT(Prop,tDewEnt)-hLiqEntSat) -- function F4 in type 75
     &       /(tDewEnt-TLiqEnt)
          CALL HEATEX(capAirWet,hAirEnt,capLiqWet,hLiqEntSat,uaH, -- subroutine S3
                                                                   in type 75
     &       ConfigHX,hAirLvg,hLiqLvgSat)
          TSurfEnt = TAIRSAT(Prop,hSurfEntSat) -- function F11 in type 75
          CALL WCOILOUT (Prop,MAir,TAirEnt,WAirEnt,hAirEnt,hAirLvg, -- subroutine S4
                                                                     in type 75
     &       UAExtTot,TAirLvg,WAirLvg,QSen,ErrStat)

       S3 SUBROUTINE HEATEX (Cap1,In1,Cap2,In2,UA,ConfigHX,Out1,Out2)
             Calculate the outlet states of a simple heat exchanger using the
             effectiveness-Ntu method of analysis.

       S4 SUBROUTINE WCOILOUT (Prop,MAir,TAirEnt,WAirEnt,HAirEnt,HAirLvg,
     &                    UAExt,TAirLvg,WAirLvg,QSen,ErrStat)
             Calculate the leaving air temperature, the leaving air humidity ratio
             and the sensible cooling capacity of wet cooling coil.
          tempCond = TAIRSAT(Prop,hCondSat) -- function F11 in type 75
          IF (tempCond .LT. DEWPOINT(Prop,WAirEnt)) THEN -- function F3 in type 75
          WAirLvg = HUMTH(Prop,TAirLvg,HAirLvg) -- function F6 in type 75
          TAirLvg = DRYBULB3(Prop,HAirLvg,WAirLvg) -- function F13 in type 75

       S5 SUBROUTINE BYPASS(Prop,TEnt,WEnt,TLvg,WLvg,
     &                 TAdp,WAdp,BF,ErrStat)
             Calculate apparatus dew point and bypass factor given entering and
             leaving moist air conditions of cooling coil.
          TAdp = DEWPOINT(Prop,WLvg) -- function F3 in type 75
          WAdp = HUMRATIO(Prop(Patm),SATPRESS(Prop,TAdp)) -- function F5 & F9
                                                             in type 75
          TAdp = XITERATE(TAdp,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                               in type 75
          hLvg = ENTHALPY3(Prop,TLvg,WLvg) -- function in F2 type 75
          hEnt = ENTHALPY3(Prop,TEnt,WEnt) -- function F2 in type 75
          hAdp = ENTHALPY3(Prop,TAdp,WAdp) -- function F2 in type 75

       F1 REAL FUNCTION UAHX (Cap1,In1,Cap2,In2,Q,ConfigHX,ErrStat)
             Calculate the UA of a heat exchanger using the effectiveness-Ntu
             relationships given the entering capacity rate and temperature of each
             flow stream, the heat transfer rate under these conditions and the heat
             exchanger configuration.
          CALL HEATEX (Cap1,In1,Cap2,In2,ua,ConfigHx,out1,out2) -- subroutine S3
                                                                  in type 75
          ua = XITERATE(ua,error,X1,F1,X2,F2,iter,icvg) -- function F14 in type 75

       F2 REAL FUNCTION ENTHALPY3 (Prop,TDB,W)
             Calculate the enthalpy of moist air.

       F3 REAL FUNCTION DEWPOINT (Prop,W)
             Calculate the dewpoint temperature given humidity ratio
          DewPoint = SATTEMP(Prop,pw) -- function F10 in type 75

       F4 REAL FUNCTION ENTHSAT (Prop,TDB)
             Calculate the enthalpy at saturation for given dry bulb temperature
          psat = SATPRESS (Prop,TDB) -- function F9 in type 75
          w = HUMRATIO (Prop(Patm),psat) -- function F5 in type 75
          ENTHSAT = ENTHALPY3 (Prop,TDB,w) -- function F2 in type 75

       F5 REAL FUNCTION HUMRATIO (Patm,Pw)
             Calculate the humidity ratio from water vapor pressure and atmospheric
             Pressure

       F6 REAL FUNCTION HUMTH (Prop,TDB,H)
             Calculate the humidity ratio of moist air from dry bulb temperature and
             enthalpy.
```

```
F7 REAL FUNCTION RELHUM (Patm,Psat,HumRatio)
      Calculate the relative humidity from saturation and atmospheric
      Pressures

F8 REAL FUNCTION RHOMOIST (RhoDry,W)
      Calculate moist air density from dry air density and humidity ratio

F9 REAL FUNCTION SATPRESS (Prop,T)
      Calculate saturation pressure of water vapor as a function of
      temperature

F10 REAL FUNCTION SATTEMP (Prop,P)
       Calculate the saturation (boiling) temperature of water given pressure
   pSat = SATPRESS(Prop,tSat) -- function F9 in type 75
   tSat = XITERATE (tSat,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                         in type 75

F11 REAL FUNCTION TAIRSAT (Prop,HSat)
       Calculate the dry bulb temperature given enthalpy at saturation.
   error = HSat - ENTHSAT(Prop,tSat) -- function F4 in type 75
   tSat = XITERATE(tSat,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                        in type 75

F12 REAL FUNCTION WETBULB (Prop,TDB,W)
       Calculate wet bulb temperature from dry bulb temperature and humidity
       ratio
   tBoil = SATTEMP (Prop,Prop(Patm)) -- function F10 in type 75
   psatStar = SATPRESS (Prop,WetBulb) -- function F9 in type 75
   wStar = HUMRATIO (Prop(Patm),psatStar) -- function F5 in type 75
   WetBulb = XITERATE(WetBulb,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                              in type 75

F13 REAL FUNCTION DRYBULB3 (Prop,H,W)
       Calculate the dry bulb temperature of moist air from enthalpy and
       humidity.

F14 REAL FUNCTION XITERATE (X0,F0,X1,F1,X2,F2,ICount,ICvg)
       Iterately solves for the value of X which satisfies F(X)=0. Given
       Xi,F(Xi) pairs, the subroutine tests for convergence and provides a new
       guess for the value of the independent variable X.
```

```
 1 PROP(PATM)     =  101325.0      Atmospheric pressure (Pa)
 2 PROP(CPAIR)    =    1006.0      Specific heat of dry air (J/kg C)
 3 PROP(CPWAT)    =    4186.0      Specific heat of liquid water (J/kg C)
 4 PROP(CPVAP)    =    1805.0      Specific heat of saturated water
                                   vapor (J/kg C)
 5 PROP(CPLIQ)    =    4186.0
 6 PROP(DVISCAIR) =        .0000182 Air dynamic viscosity (kg/m s)
 7 PROP(DVISCLIQ) =        .00144  Liquid dynamic viscosity (kg/m s)
 8 PROP(KAIR)     =        .026    Air thermal conductivity (W/m C)
 9 PROP(KLIQ)     =        .604    Liquid thermal conductivity (W/m C)
10 PROP(RHOLIQ)   =     998.0      Liquid density (kg/m3)
11 PROP(HFG)      = 2501000.0      Latent heat of vaporization of water (J/kg)
12 PROP(RAIR)     =     287.055    Gas constant for air (J/kg C)
13 PROP(TKELMULT) =       1.0      Multiplying factor to convert user
                                   T to Kelvin
14 PROP(TABSADD)  =     273.15     Additive factor to convert user P to Kelvin:
                                   tKel = Prop(TKelMult)*T + Prop(TKelAdd)
15 PROP(PAMULT)   =       1.0      Multiplying factor to convert user P to
                                   Pascals
16 PROP(PABSADD)  =       0.0      Additive factor to convert user P to Pascals:
                                   Pa = Prop(PaMult)*P + Prop(PaAdd)
```

SOURCE CODE

```
      SUBROUTINE TYPE75 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C********************************************************************
C*    SUBROUTINE: CCSIMinverted
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Model the performance of a counterflow
C*                crossflow cooling coil. The model accounts
C*                for condensation on the outside surface.
C*                Three conditions are possible:  all wet,
C*                partially wet or all dry. Input includes
C*                outlet air setpoint temperature. Water flow
C*                rate is changed until desired value is
C*                achieved. Output includes
C*                outlet air temperature and humidity, outlet
C*                water temperature, sensible and total
C*                cooling capacities and the wet fraction of
C*                air-side surface area.
C*******************************************************************
C*    INPUT VARIABLES  DESCRIPTION(UNITS)                        SAMPLE VALUES
C*    XIN(1)     MLiq Liquid mass flow rate(kg/s)                     4.2
C*    XIN(2)  TLiqEnt Entering water temperature(C)                   5.5556
C*    XIN(3)     MAir Dry air mass flow rate(kg/s)                    3.2
C*    XIN(4)  TAirEnt Entering air dry bulb temperature(C)           25.0
C*    XIN(5)  WAirEnt Entering air humidity ratio(-)                   .01
C*    XIN(6)  Tsp            Leaving air setpoint temperature(C)     11.0
C*
C*    OUTPUT VARIABLES
C*    OUT(1)  TLiqLvg Leaving water temperature(C)                    9.16554
C*    OUT(2)     Mlig Liquid mass flow rate(kg/s)                     4.2
C*    OUT(3)  TAirLvg Leaving air dry bulb temperature(C)            11.0299
C*    OUT(4)  WAirLvg Leaving air humidity ratio(-)                    .0078074
C*    OUT(5)     QTot Total heat transfer rate(W)                  63467.1
C*    OUT(6)     QSen Sensible heat transfer rate(W)               45779.5
C*    OUT(7)     FWet Fraction of surface area wet(-)                 1.0
C*    OUT(8)  HAirent Enthalpy of entering air (J/kg)              30000.
C*    OUT(9)  ErrStat Error status indicator,0=ok,1=error(-)          0.0
C*
C*    PARAMETERS
C*    PAR(1)  MLiqRat Liquid mass flow rate at rating(kg/s)           4.2
C*    PAR(2)  TLiqRat Entering water temperature at rating(C)         5.5556
C*    PAR(3)  MAirRat Dry air mass flow rate at rating(kg/s)          6.4
C*    PAR(4)  TAirRat Entering air dry bulb temperature at rating(C) 26.6667
C*    PAR(5)  WAirRat Entering air humidity ratio at rating(-)         .0112
C*    PAR(6)  QTotRat Total heat transfer rate at rating(W)        88000.0
C*    PAR(7)  QSenRat Sensible heat transfer rate at rating(W)     66000.0
C*
C*    PROPERTIES
C*    CpAir         Dry air specific heat                         (J/kg C)
C*    CpVap         Water vapor specific heat                     (J/kg C)
C*    CpLiq         Liquid specific heat                          (J/kg C)
C*******************************************************************
C     MAJOR RESTRICTIONS:     General application is for heat exchanger
C                             with four or more rows in a counterflow
C                             configuration.
C                             Approximates part-wet operation as
C                             either fully wet or fully dry.
C                             Constant UA.
C
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
```

```
C
C     INCLUDE FILES:          coilsim.inc
C                             prop.inc
C     SUBROUTINES CALLED:     DRYCOIL
C                             WETCOIL
C                             HEATEX
C                             UAHX
C                             WCOILOUT
C                             BYPASS
C                             XITERATE
C     FUNCTIONS CALLED:       DEWPOINT
C                             DRYBULB3
C                             ENTHALPY3
C                             ENTHSAT
C                             HUMTH
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              TRNSYS.  1990.  A Transient System
C                             Simulation Program: Reference Manual.
C                             Solar Energy Laboratory, Univ. Wisconsin-
C                             Madison, pp. 4.6.8-1 - 4.6.8-12.
C
C                             Threlkeld, J.L.  1970.  Thermal
C                             Environmental Engineering, 2nd Edition,
C                             Englewood Cliffs: Prentice-Hall,Inc.
C                             pp. 254-270.
C*********************************************************************
C     INTERNAL VARIABLES
C     P(UAExt)      Overall external dry UA/total external area    (W/C)
C     P(UAInt)      Overall internal UA/total external area        (W/C)
C     P(UATot)      Overall heat transfer coefficient              (W/C)
C     uaH           Enthalpy-based overall transfer coefficient   (kg/s)
C     configHX      Heat exchanger configuration                     (-)
C     hAirRat       Entering air enthalpy at rating               (J/kg)
C     hAirLvgRat    Leaving air enthalpy at rating                (J/kg)
C     hAdpRat       Air enthalpy at apparatus dew point at rating (J/kg)
C     hLiqRatSat    Saturated enthalpy at entering liquid temp    (J/kg)
C     tAirLvgRat    Leaving air temperature at rating                (C)
C     tDewRat       Entering air dewpoint at rating                  (C)
C     tSurfEnt      Coil surface temperature at air entrance         (C)
C     capAir        Air-side capacity rate                         (W/C)
C     capAirH       Enthalpy-based air-side capacity rate         (kg/s)
C     capLiqH       Enthalpy-based liquid-side capacity rate      (kg/s)
C     small         Small number in place of zero
C     large         Large number in place of infinity
C*********************************************************************
      DOUBLE PRECISION XIN,OUT

      DIMENSION XIN(6),OUT(9),PAR(7),INFO(15)

      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,ERRSTAT,
     &        iter,itmax

      REAL Prop(16),LARGE,UATOT,UAINT,UAEXT,PAR,MLIQ,MAIR,
     &     Tsp,error,Hairent,Mairrat,Mliqrat

      INTEGER INFO,IOPT,NI,NP,ND

      CHARACTER*3 YCHECK(6),OCHECK(9)

      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      PARAMETER (Patm     = 1)
```

```
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      PROP(PATM)     =  101325.0
      PROP(CPAIR)    =    1006.0
      PROP(CPWAT)    =    4186.0
      PROP(CPVAP)    =    1805.0
      PROP(CPLIQ)    =    4186.0
      PROP(DVISCAIR) =        .0000182
      PROP(DVISCLIQ) =        .00144
      PROP(KAIR)     =        .026
      PROP(KLIQ)     =        .604
      PROP(RHOLIQ)   =     998.0
      PROP(HFG)      = 2501000.0
      PROP(RAIR)     =     287.055
      PROP(TKELMULT) =       1.0
      PROP(TABSADD)  =     273.15
      PROP(PAMULT)   =       1.0
      PROP(PABSADD)  =       0.0

      DATA small/1.E-9/, large /1.E20/, configHX /1./
      DATA YCHECK/'MF2','TE1','MF2','TE1','DM1','TE1'/
      DATA OCHECK/'TE1','MF2','TE1','DM1','PW2','PW2','DM1','PW2','DM1'/
      DATA itmax/50/

      ErrStat = 0
      IOPT    = -1
      NI      = 6             !CORRECT NUMBER OF INPUTS
      NP      = 7             !CORRECT NUMBER OF PARAMETERS
      ND      = 0             !CORRECT NUMBER OF DERIVATIVES

      MLIQ    = XIN(1)
      TLIQENT = XIN(2)
      MAIR    = XIN(3)
      TAIRENT = XIN(4)
      WAIRENT = XIN(5)
      TSP     = XIN(6)

      MLIQRAT = PAR(1)
      TLIQRAT = PAR(2)
      MAIRRAT = PAR(3)
      TAIRRAT = PAR(4)
      WAIRRAT = PAR(5)
      QTOTRAT = PAR(6)
      QSENRAT = PAR(7)

      IF (INFO(7).EQ.-1) THEN
            CALL TYPECK(IOPT,INFO,NI,NP,ND)
C       CHECKS TO SEE IF USER'S INFO MATCHES CORRECT NUMBERS
            CALL RCHECK(INFO,YCHECK,OCHECK)
C       CHECKS TO SEE IF INPUT AND OUTPUT UNITS MATCH
          INFO(6)=9
      ENDIF
```

```
C1*** If economizer meets supply air temp setpoint then no coil load
      If (TSP .GE. TAIRENT) THEN
        TLIQLVG=TLIQENT
        MLIQ=0.
        TAIRLVG=TAIRENT
        WAIRLVG=WAIRENT
        QTOT=0.
        QSEN=0.
        FWET=0.
        GOTO 999
      ENDIF

C2*****************************************************************
C2    The code between these bars of asterisks is used to set internal
C2    parameters and is independent of component input values.  In an
C2    hourly simulation, this block of code may be skipped after the
C2    first call.

C1*** Calculate properties of air and liquid at rating point
      hAirRat = ENTHALPY3(Prop,TAirRat,WAirRat)
      hLiqRatSat = ENTHSAT(Prop,TLiqRat)

C1*** Calculate leaving air states at rating point
      hAirLvgRat = hAirRat - QTotRat/MAirRat
      hDummy = hAirRat - (QTotRat-QSenRat)/MAirRat
      wAirLvgRat = HUMTH(Prop,TAirRat,hDummy)
      tAirLvgRat = DRYBULB3(Prop,hAirLvgRat,wAirLvgRat)

C1*** Calculate coil UA assuming wet coil at rating

C2***   Heat transfer in a wet coil is calculated based on enthalpy
C2***   rather than temperature to include latent effects.  Air enthalpies
C2***   are evaluated using conventional psychrometric equations.  The
C2***   corresponding enthalpies of the coil and water are related to
C2***   that of the air through "fictitious enthalpies," defined as the
C2***   enthalpy of saturated air at the temperature of the coil or water.
C2
C2***   While heat transfer rates are commonly expressed as the product
C2***   of an overall heat transfer coefficient, UA, and a temperature
C2***   difference, the use of enthalpy-based heat transfer calculations
C2***   requires an enthalpy-based heat transfer coefficient, UAH.
C2
C2***       q = UAH * (H1-H2)
C2
C2***   where UAH = UA / cp
C2***         UA = conventional heat transfer coefficient
C2***         cp = specific heat across enthalpy difference
C2
C2***   When using fictitious enthalpies, a corresponding fictitious
C2***   specific heat must be defined.
C2
C2***       EnthSat1-EnthSat2 = cpSat * (Temp1-Temp2)
C2
C2***   UAH can be calculated from a combination of series or parallel
C2***   enthalpy resistances, similar to thermal resistances modified for
C2***   enthalpy as above.  Enthalpy capacity rates relate heat transfer
C2***   to the enthalpy change of a fluid between inlet and outlet.
C2
C2***       q = CapH * (HAirLvg - HAirEnt)
C2
C2***   On the air side, enthalpy capacity rate is the air mass flow rate.
C2***   On the water side, the enthalpy capacity rate is based on the
C2***   enthalpy of saturated air at the water temperature.

C1*** Estimate cpSat using entering air dewpoint and water temperature
```

```
      tDewRat = DEWPOINT(Prop,WAirRat)
      cpSat = (ENTHSAT(Prop,tDewRat)-hLiqRatSat)
     &        /(tDewRat-TLiqRat)

C1*** Calculate overall heat transfer coefficient from fluid states
C1*** and known total heat transfer
      capAirH = MAirRat
      capLiqH = MLiqRat * (Prop(CpLiq)/cpSat)
      uaH = UAHX(capAirH,hAirRat,capLiqH,hLiqRatSat,QTotRat,
     &            configHX,ErrStat)

C1*** Determine air-side coefficient, UAExt, assuming that the
C1*** surface temperature is at the apparatus dewpoint temperature
      CALL BYPASS(Prop,TairRat,WAirRat,tAirLvgRat,wAirLvgRat,
     &            tAdpRat,wAdpRat,bfRat,ErrStat)
      hAdpRat = ENTHALPY3(Prop,tAdpRat,wAdpRat)

      IF (hAdpRat .LE. hLiqRatSat) THEN
        UAExt = uaH*Prop(CpAir)
      ELSE
        capAir = MAirRat*(Prop(CpAir)+WAirRat*Prop(CpVap))
        UAExt = -LOG(bfRat)*capAir
      ENDIF

C1*** Calculate liquid-side coefficient, UAInt, from enthalpy-based
C1*** overall coefficient and air-side coefficient
      UAInt = cpSat/MIN((1./uaH - Prop(CpAir)/UAExt),large)
      UATot = 1./(1./UAExt+1./UAInt)

C2*********************************************************************

C1*** If both flows are zero, set outputs to inputs and return

      IF (ABS(MAir) .LT. small .AND. ABS(MLiq) .LT. small) THEN
        TLiqLvg = TLiqEnt
        TAirLvg = TAirEnt
        WAirLvg = WAirEnt
        GO TO 999
      ENDIF

C1*** BEGIN LOOP
      DO 100 iter = 1 ,itmax
C1*** IF coil is completely dry THEN
        tDewPt = DEWPOINT (Prop,WAirEnt)
        IF (tDewPt .LE. TLiqEnt) THEN
C1*** Calculate the leaving conditions and performance of dry coil
          CALL DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                  UATot,configHX,
     &                  TLiqLvg,TAirLvg,WAirLvg,QTot,ErrStat)
          QSen = QTot
          FWet = 0.
        ELSE
C1*** ELSE Assume external surface of coil is completely wet
C1*** Calculate the leaving conditions and performance of wet coil
          CALL WETCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                  UAInt,UAExt,configHX,
     &                  TLiqLvg,TAirLvg,WAirLvg,QTot,QSen,FWet,
     &                  tSurfEnt,ErrStat)
C1*** IF coil is only partially wet THEN
C       IF (tDewPt .LT. tSurfEnt) THEN
C1*** Calculate the leaving conditions and performance of dry coil
C         CALL DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
C    &                UATot,configHX,
C    &                dryTLiqLvg,dryTAirLvg,dryWAirLvg,dryQTot,ErrStat)
C1*** IF heat transfer from drycoil calculations is greater than that
C1*** from wetcoil calculations THEN approximate the coil as dry.
```

```
C         IF (dryQTot .GT. QTot) THEN
C           TLiqLvg = dryTLiqLvg
C           TAirLvg = dryTAirLvg
C           WAirLvg = dryWAirLvg
C           QTot = dryQTot
C            QSen = QTot
C            FWet = 0.
C         ENDIF
C       ENDIF
        ENDIF
C1*** Compare given leaving air temperature with estimated temperature
C1*** and determine new estimate of flow
      error = TAirLvg-Tsp
      mLiq = XITERATE(mLiq,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave loop
        IF (icvg .EQ. 1) GO TO 999
C1*** If estimated flow is less than zero, set to small number
        IF(MLiq.LT.0) MLiq = 0.
 100  CONTINUE

C1*** If not converged after itmax iterations, return error code
      WRITE(LUW,1005) itmax
 1005 FORMAT(/1X,'*** ERROR IN SUBROUTINE COILINV ***'/
     &        1X,'    Temperature has not converged after',I2,
     &        '  iterations'/)
      ErrStat = 1

999   CONTINUE

      hAirEnt = MAIR * ENTHALPY3(Prop,TAirEnt,WAirEnt)

      OUT(1) = TLIQLVG
      OUT(2) = MLIQ
      OUT(3) = TAIRLVG
      OUT(4) = WAIRLVG
      OUT(5) = QTOT
      OUT(6) = QSEN
      OUT(7) = FWET
      OUT(8) = hAirEnt
      OUT(9) = ERRSTAT

      RETURN 1
      END

C***********************************************************************
C*    FILE: PROP.INC
C*
C*    This file assigns a numbers to air and water property names to be
C*    used in the "Prop" array.
C***********************************************************************
C     DEVELOPER:         Inger Andresen
C                        Michael J. Brandemuehl, PhD, PE
C
C     DATE:              July 1, 1991
C
C     FILES REQUIRED:    None
C***********************************************************************
C     INTERNAL VARIABLES:
C     Patm          Atmospheric pressure                            (Pa)
C     CpAir         Specific heat of dry air                    (J/kg C)
C     CpLiq         Specific heat of liquid water               (J/kg C)
C     CpVap         Specific heat of saturated water vapor      (J/kg C)
C     DViscAir      Air dynamic viscosity                       (kg/m s)
C     DViscLiq      Liquid dynamic viscosity                    (kg/m s)
C     KAir          Air thermal conductivity                     (W/m C)
C     KLiq          Liquid thermal conductivity                  (W/m C)
```

```
C     RhoLiq        Liquid density                                (kg/m3)
C     Hfg           Latent heat of vaporization of water           (J/kg)
C     RAir          Gas constant for air                         (J/kg C)
C     TKelMult      Multiplying factor to convert user T to Kelvin
C     TAbsAdd       Additive factor to convert user P to Kelvin
C                   tKel = Prop(TKelMult)*T + Prop(TKelAdd)
C     PaMult        Multiplying factor to convert user P to Pascals
C     PAbsAdd       Additive factor to convert user P to Pascals
C                   Pa = Prop(PaMult)*P + Prop(PaAdd)
C*********************************************************************
C
C      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
C     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
C     &        TKelMult,TAbsAdd,PaMult,PAbsAdd
C      REAL Prop(16)
C
C      PARAMETER (Patm     = 1)
C      PARAMETER (CpAir    = 2)
C      PARAMETER (CpWat    = 3)
C      PARAMETER (CpVap    = 4)
C      PARAMETER (CpLiq    = 5)
C      PARAMETER (DViscAir = 6)
C      PARAMETER (DViscLiq = 7)
C      PARAMETER (KAir     = 8)
C      PARAMETER (KLiq     = 9)
C      PARAMETER (RhoLiq   = 10)
C      PARAMETER (Hfg      = 11)
C      PARAMETER (RAir     = 12)
C      PARAMETER (TKelMult = 13)
C      PARAMETER (TAbsAdd  = 14)
C      PARAMETER (PaMult   = 15)
C      PARAMETER (PAbsAdd  = 16)

      SUBROUTINE DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                    UA,ConfigHX,
     &                    TLiqLvg,TAirLvg,WAirLvg,Q,
     &                    ErrStat)
C*********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*********************************************************************
C*    SUBROUTINE: DRYCOIL
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the performance of a sensible
C*                air-liquid heat exchanger.  Calculated
C*                results include outlet air temperature
C*                and humidity, outlet water temperature,
C*                and heat transfer rate.
C*********************************************************************
C*    INPUT VARIABLES
C*    MLiq          Liquid mass flow rate                          (kg/s)
C*    TLiqEnt       Entering water temperature                        (C)
C*    MAir          Dry air mass flow rate                         (kg/s)
C*    TAirEnt       Entering air dry bulb temperature                 (C)
C*    WAirEnt       Entering air humidity ratio                       (-)
C*
C*    UA            Overall heat transfer coefficient               (W/C)
C*    ConfigHX      Heat exchanger configuration                      (-)
C*                  1 - Counterflow
C*                  2 - Parallel flow
C*                  3 - Cross flow, both streams unmixed
C*                  4 - Cross flow, both streams mixed
C*                  5 - Cross flow, stream 1 unmixed
C*                  6 - Cross flow, stream 2 unmixed
C*
```

```
C*    OUTPUT VARIABLES
C*    TLiqLvg       Leaving water temperature                       (C)
C*    TAirLvg       Leaving air dry bulb temperature                (C)
C*    WAirLvg       Leaving air humidity ratio                      (-)
C*    Q             Heat transfer rate                              (W)
C*    ErrStat       Error status indicator, 0 = ok, 1 = error       (-)
C*
C*    PROPERTIES
C*    CpAir         Specific heat of air                       (J/kg C)
C*    CpVap         Specific heat of water vapor               (J/kg C)
C*    CpLiq         Specific heat of liquid                    (J/kg C)
C*******************************************************************
C     MAJOR RESTRICTIONS:     Models coil using effectiveness-Ntu model.
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     HEATEX
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              Kays, W.M. and A.L. London.  1964.
C                             Compact Heat Exchangers, 2nd Edition,
C                             New York: McGraw-Hill.
C
C                             Threlkeld, J.L.  1970.  Thermal
C                             Environmental Engineering, 2nd Edition,
C                             Englewood Cliffs: Prentice-Hall, Inc.
C                             pp. 254-270.
C*******************************************************************
C     INTERNAL VARIABLES:
C     capAir        Air-side capacity rate                         (W/C)
C     capLiq        Water-side capacity rate                       (W/C)
C*******************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      REAL MAir,MLiq

      INTEGER Errstat
```

```
      Errstat = 0

C2*** Calculate air and water capacity rates
      capAir = MAir*(Prop(CpAir)+WAirEnt*Prop(CpVap))
      capLiq = MLiq*Prop(CpLiq)

C1*** Determine the air and water outlet conditions
      CALL HEATEX (capLiq,TLiqEnt,capAir,TAirEnt,UA,ConfigHX,
     &             TLiqLvg,TAirLvg)

C1*** Calculate the total and sensible heat transfer rate
      Q = capAir*(TAirEnt-TAirLvg)
      WAirLvg = WAirEnt

      RETURN
      END

      SUBROUTINE WETCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,
     &                    UAIntTot,UAExtTot,ConfigHX,
     &                    TLiqLvg,TAirLvg,WAirLvg,QTot,QSen,FWet,
     &                    TSurfEnt,ErrStat)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    SUBROUTINE: WETCOIL
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the performance of a cooling
C*                coil when the external fin surface is
C*                complete wet.  Results include
C*                outlet air temperature and humidity,
C*                outlet water temperature, sensible and
C*                total cooling capacities, and the wet
C*                fraction of the air-side surface area.
C***********************************************************************
C*    INPUT VARIABLES
C*    MLiq          Liquid mass flow rate                          (kg/s)
C*    TLiqEnt       Entering water temperature                        (C)
C*    MAir          Dry air mass flow rate                         (kg/s)
C*    TAirEnt       Entering air dry bulb temperature                 (C)
C*    WAirEnt       Entering air humidity ratio                       (-)
C*
C*    UAIntTot      Internal overall heat transfer coefficient  (W/m2 C)
C*    UAExtTot      External overall heat transfer coefficient  (W/m2 C)
C*    ConfigHX      Heat exchanger configuration                      (-)
C*                  1 - Counterflow
C*                  2 - Parallel flow
C*                  3 - Cross flow, both streams unmixed
C*                  4 - Cross flow, both streams mixed
C*                  5 - Cross flow, stream 1 unmixed
C*                  6 - Cross flow, stream 2 unmixed
C*
C*    OUTPUT VARIABLES
C*    TLiqLvg       Leaving water temperature                         (C)
C*    TAirLvg       Leaving air dry bulb temperature                  (C)
C*    WAirLvg       Leaving air humidity ratio                        (-)
C*    QTot          Total heat transfer rate                          (W)
C*    QSen          Sensible heat transfer rate                       (W)
C*    FWet          Fraction of surface area wet                      (-)
C*    TSurfEnt      Surface temperature at air entrance               (C)
C*    ErrStat       Error status indicator, 0 = ok, 1 = error         (-)
C*
C*    PROPERTIES
C*    CpLiq         Specific heat of liquid                     (J/kg C)
C*    CpAir         Specific heat of dry air                    (J/kg C)
```

```
C*******************************************************************
C     MAJOR RESTRICTIONS:     Models coil as counterflow heat exchanger
C                             Approximates saturated air enthalpy as
C                             a linear function of temperature
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     HEATEX
C                             WCOILOUT
C     FUNCTIONS CALLED:       ENTHALPY3
C                             ENTHSAT
C                             TAIRSAT
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              Elmahdy, A.H. and Mitalas, G.P.  1977.
C                             "A Simple Model for Cooling and
C                             Dehumidifying Coils for Use In Calculating
C                             Energy Requirements for Buildings,"
C                             ASHRAE Transactions,Vol.83 Part 2,
C                             pp. 103-117.
C
C                             TRNSYS.  1990.  A Transient System
C                             Simulation Program: Reference Manual.
C                             Solar Energy Laboratory, Univ. Wisconsin-
C                             Madison, pp. 4.6.8-1 - 4.6.8-12.
C
C                             Threlkeld, J.L.  1970.  Thermal
C                             Environmental Engineering, 2nd Edition,
C                             Englewood Cliffs: Prentice-Hall,Inc.
C                             pp. 254-270.
C*******************************************************************
C     INTERNAL VARIABLES:
C     extResist     Air-side resistance to heat transfer        (m2 C/W)
C     intResist     Liquid-side resistance to heat transfer     (m2 C/W)
C     tDewEnt       Entering air dew point                          (C)
C     uaH           Overall enthalpy heat transfer coefficient   (kg/s)
C     capAirWet     Air-side capacity rate                       (kg/s)
C     capLiqWet     Liquid-side capacity rate                    (kg/s)
C     resistRatio   Ratio of resistances                            (-)
C     hAirLvg       Outlet air enthalpy
C     hLiqEntSat    Saturated enthalpy of air at                 (J/kg)
C                   entering water temperature
C     hLiqLvgSat    Saturated enthalpy of air at exit            (J/kg)
C                   water temperature
C     hSurfEntSat   Saturated enthalpy of air at                 (J/kg)
C                   entering surface temperature
C     hSurfLvgSat   Saturated enthalpy of air at exit            (J/kg)
C                   surface temperature
C     cpSat         Coefficient for equation below             (J/kg C)
C                   EnthSat1-EnthSat2 = cpSat*(TSat1-TSat2)
C                   (all water and surface temperatures are
C                   related to saturated air enthalpies for
C                   wet surface heat transfer calculations)
C********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)
```

```
      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      REAL MAir,MLiq,intResist

      INTEGER ErrStat

      DATA small/1.E-9/

      FWet = 1.
      extResist = 1./UAExtTot
      intResist = 1./UAIntTot

C1*** Calculate enthalpies of entering air and water
      hAirEnt = ENTHALPY3(Prop,TAirEnt,WAirEnt)
      hLiqEntSat = ENTHSAT(Prop,TLiqEnt)

C1*** Estimate cpSat using entering air dewpoint and water temperature
      tDewEnt = DEWPOINT(Prop,WAirEnt)
      cpSat = (ENTHSAT(Prop,tDewEnt)-hLiqEntSat)
     &        /(tDewEnt-TLiqEnt)

C1*** Enthalpy-based heat transfer calculations

C2***   Heat transfer in a wet coil is calculated based on enthalpy
C2***   rather than temperature to include latent effects.  Air enthalpies
C2***   are evaluated using conventional psychrometric equations.  The
C2***   corresponding enthalpies of the coil and water are related to
C2***   that of the air through "fictitious enthalpies," defined as the
C2***   enthalpy of saturated air at the temperature of the coil or water.
C2
C2***   While heat transfer rates are commonly expressed as the product
C2***   of an overall heat transfer coefficient, UA, and a temperature
C2***   difference, the use of enthalpy-based heat transfer calculations
C2***   requires an enthalpy-based heat transfer coefficient, UAH.
C2
C2***       q = UAH * (H1-H2)
C2
C2***   where UAH = UA / cp
C2***         UA = conventional heat transfer coefficient
C2***         cp = specific heat across enthalpy difference
C2
C2***   When using fictitious enthalpies, a corresponding fictitious
C2***   specific heat must be defined.
C2
C2***       EnthSat1-EnthSat2 = cpSat * (Temp1-Temp2)
C2
C2***   UAH can be calculated from a combination of series or parallel
C2***   enthalpy resistances, similar to thermal resistances modified for
C2***   enthalpy as above.  Enthalpy capacity rates relate heat transfer
C2***   to the enthalpy change of a fluid between inlet and outlet.
C2
```

```
C2***       q = CapH * (HAirLvg - HAirEnt)
C2
C2***   On the air side, enthalpy capacity rate is the air mass flow rate.
C2***   On the water side, the enthalpy capacity rate is based on the
C2***   enthalpy of saturated air at the water temperature.

C1*** Determine air and water enthalpy outlet conditions by modeling
C1*** coil as counterflow enthalpy heat exchanger
        uaH = 1./(cpSat*intResist+Prop(CpAir)*extResist)
        capAirWet = MAir
        capLiqWet = MLiq * (Prop(CpLiq)/cpSat)
        CALL HEATEX(capAirWet,hAirEnt,capLiqWet,hLiqEntSat,uaH,
     &             ConfigHX,hAirLvg,hLiqLvgSat)

C1*** Calculate entering and leaving external surface conditions from
C1*** air and water conditions and the ratio of resistances
        resistRatio = (intResist)/(intResist +
     &               Prop(CpAir)/cpSat*extResist)
        hSurfEntSat = hLiqLvgSat + resistRatio*(hAirEnt-hLiqLvgSat)
        hSurfLvgSat = hLiqEntSat + resistRatio*(hAirLvg-hLiqEntSat)
        TSurfEnt = TAIRSAT(Prop,hSurfEntSat)

C1*** Calculate outlet air temperature and humidity from enthalpies and
C1*** surface conditions.
      QTot = MAir*(hAirEnt-hAirLvg)
      TLiqLvg = TLiqEnt+QTot/MAX(MLiq,small)/Prop(CpLiq)
      CALL WCOILOUT (Prop,MAir,TAirEnt,WAirEnt,hAirEnt,hAirLvg,
     &               UAExtTot,TAirLvg,WAirLvg,QSen,ErrStat)

  999 RETURN
      END

      SUBROUTINE HEATEX (Cap1,In1,Cap2,In2,UA,ConfigHX,Out1,Out2)
C*********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*********************************************************************
C*    SUBROUTINE: HEATEX
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the outlet states of a simple
C*                heat exchanger using the effectiveness-Ntu
C*                method of analysis.
C*********************************************************************
C*    INPUT VARIABLES
C*    Cap1          Capacity rate of stream 1                     (W/C)
C*    In1           Inlet state of stream 1                         (C)
C*    Cap2          Capacity rate of stream 2                     (W/C)
C*    In2           Inlet state of stream 2                         (C)
C*    UA            Overall heat transfer coefficient             (W/C)
C*    ConfigHX      Heat exchanger configuration                    (-)
C*                  1 - Counterflow
C*                  2 - Parallel flow
C*                  3 - Cross flow, both streams unmixed
C*                  4 - Cross flow, both streams mixed
C*                  5 - Cross flow, stream 1 unmixed
C*                  6 - Cross flow, stream 2 unmixed
C*
C*    OUTPUT VARIABLES
C*    Out1          Outlet state of stream 1                        (C)
C*    Out2          Outlet state of stream 2                        (C)
C*********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
```

```
C                              University of Colorado at Boulder
C
C     DATE:                    January 1, 1992
C
C     INCLUDE FILES:           None
C     SUBROUTINES CALLED:      None
C     FUNCTIONS CALLED:        None
C
C     REVISION HISTORY:        None
C
C     REFERENCE:               Kays, W.M. and A.L. London.  1964.
C                              Compact Heat Exchangers, 2nd Ed., McGraw-
C                              Hill:  New York.
C***********************************************************************
C*    INTERNAL VARIABLES:
C*    cMin          Minimum capacity rate of the streams          (W/C)
C*    cMax          Maximum capacity rate of the streams          (W/C)
C*    cRatio        Ratio of minimum to maximum capacity rate
C*    ntu           Number of transfer units                        (-)
C*    effectiveness Heat exchanger effectiveness                    (-)
C*    qMax          Maximum heat transfer possible                  (W)
C***********************************************************************
      REAL ntu,qMax,In1,In2,large

      DATA small/1.E-15/, large/1.E15/

C1*** Ntu and Cmin/Cmax (cRatio) calculations
      cMin = MIN(Cap1,Cap2)
      cMax = MAX(Cap1,Cap2)

      IF( cMax .EQ. 0.) THEN
        cRatio = 1.
      ELSE
        cRatio = cMin/cMax
      ENDIF

      IF( cMin .EQ. 0.) THEN
        ntu = large
      ELSE
        ntu = ua/cMin
      ENDIF

C1*** Calculate effectiveness for special limiting cases
      mode = NINT(ConfigHX)
      IF(ntu .LE. 0) THEN
        effectiveness = 0.
      ELSE IF(cRatio .LT. small) THEN
C2*** Cmin/Cmax = 0 and effectiveness is independent of configuration
        effectiveness = 1 - EXP(-ntu)
C1*** Calculate effectiveness depending on heat exchanger configuration
      ELSE IF (mode .EQ. 1) THEN
C2*** Counterflow
        IF (ABS(cRatio-1.) .LT. small) THEN
          effectiveness = ntu/(ntu+1.)
        ELSE
          e=EXP(-ntu*(1-cRatio))
          effectiveness = (1-e)/(1-cRatio*e)
        ENDIF
      ELSE IF (mode .EQ. 2) THEN
C2*** Parallel flow
        effectiveness = (1-EXP(-ntu*(1+cRatio)))/(1+cRatio)
      ELSE IF (mode .EQ. 3) THEN
C2*** Cross flow, both streams unmixed
        eta = ntu**(-0.22)
        effectiveness = 1 - EXP((EXP(-ntu*cRatio*eta)-1)/(cRatio*eta))
      ELSE IF (mode .EQ. 4) THEN
```

```
C2*** Cross flow, both streams mixed
        effectiveness = ((1/(1-EXP(-ntu))+
     &                 (cRatio/(1-EXP(-ntu*cRatio)))-(1/(-ntu)))**(-1)
      ELSE
C2*** One stream is mixed and one is unmixed.  Determine whether the
C2*** minimum or maximum capacity rate stream is mixed.
        IF ( (ABS(Cap1-cMin).LT.small .AND. mode.EQ.5) .OR.
     &       (ABS(Cap2-cMin).LT.small .AND. mode.EQ.6) ) THEN
C2*** Cross flow, stream with minimum capacity rate unmixed
          effectiveness = (1-EXP(-cRatio*(1-EXP(-ntu))))/cRatio
        ELSE
C2*** Cross flow, stream with maximum capacity rate unmixed
          effectiveness = 1-EXP(-(1-EXP(-ntu*cRatio))/cRatio)
        ENDIF
      ENDIF

C1*** Determine leaving conditions for the two streams
      qMax = MAX(cMin,small)*(In1-In2)
      Out1 = In1 - effectiveness*qMax/MAX(Cap1,small)
      Out2 = In2 + effectiveness*qMax/MAX(Cap2,small)

      RETURN
      END

      REAL FUNCTION UAHX (Cap1,In1,Cap2,In2,Q,ConfigHX,ErrStat)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: UAHX
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the UA of a heat exchanger
C*              using the effectiveness-Ntu relationships
C*              given the entering capacity rate and
C*              temperature of each flow stream, the
C*              heat transfer rate under these conditions
C*              and the heat exchanger configuration.
C***********************************************************************
C*    INPUT VARIABLES
C*    Cap1          Capacity rate of stream 1                     (W/C)
C*    In1           Inlet state of stream 1                         (C)
C*    Cap2          Capacity rate of stream 2                     (W/C)
C*    In2           Inlet state of stream 2                         (C)
C*    Q             Heat transfer rate                              (W)
C*    ConfigHX      Heat exchanger configuration                    (-)
C*                  1 - Counterflow
C*                  2 - Parallel flow
C*                  3 - Cross flow, both streams unmixed
C*                  4 - Cross flow, both streams mixed
C*                  5 - Cross flow, stream 1 unmixed
C*                  6 - Cross flow, stream 2 unmixed
C*
C*    OUTPUT VARIABLES
C*    UAHX          Overall heat transfer coefficient             (W/C)
C*    ErrStat       Error status indicator, 0 = ok, 1 = error       (-)
C***********************************************************************
C     MAJOR RESTRICTIONS:     Models coil using effectiveness Ntu model
C
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     HEATEX
```

```
C     FUNCTIONS CALLED:       XITERATE
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              None
C*********************************************************************
C     INTERNAL VARIABLES
C     qEstimate     Estimated heat transfer in iteration             (W)
C     ua            Estimated heat transfer coefficient            (W/C)
C     error         Deviation of dependent variable in iteration
C     icvg          Iteration convergence flag
C     iter          Iteration index
C     itmax         Maximum number of iterations
C     F1,F2         Previous values of error in iteration
C     X1,X2         Previous values of independent variable in iteration
C*********************************************************************
      REAL In1,In2

      INTEGER ErrStat

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA itmax/20/

      ErrStat = 0

C1*** Check for Q out of range (effectiveness > 1)
      IF(ABS(Q) .GT. ABS(MIN(Cap1,Cap2)*(In1-In2))) THEN
        WRITE(LUW,1001)
 1001   FORMAT(/1X,'*** ERROR IN SUBROUTINE UAHX ***'/
     &          1X,'    Given Q is impossible for given inlet states'/)
        ErrStat = 1
      ENDIF

C1*** Estimate UAHX
      ua = ABS(Q/(In1-In2))

C1*** BEGIN LOOP to iteratively calculate UAHX
      DO 100 iter = 1,itmax
C1*** Calculate heat transfer rate for estimated UAHX
        CALL HEATEX (Cap1,In1,Cap2,In2,ua,ConfigHx,out1,out2)
        qEstimate = Cap1*(In1-out1)
C1*** Calculate new estimate for UAHX
        error = ABS(qEstimate) - ABS(Q)
        ua = XITERATE(ua,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave loop
        IF (icvg .EQ. 1) GO TO 110
  100 CONTINUE

C1*** If not converged after itmax iterations, return error code
      WRITE(LUW,1005) itmax
 1005 FORMAT(/1X,'*** ERROR IN SUBROUTINE UAHX ***'/
     &        1X,'    UA has not converged after',I2,
     &        '  iterations'/)
      ErrStat = 1

  110 CONTINUE

      UAHX = ua

      RETURN
      END

      SUBROUTINE WCOILOUT (Prop,MAir,TAirEnt,WAirEnt,HAirEnt,HAirLvg,
     &                     UAExt,TAirLvg,WAirLvg,QSen,ErrStat)
```

```
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*    SUBROUTINE: WCOILOUT
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the leaving air temperature,
C*                the leaving air humidity ratio and the
C*                sensible cooling capacity of wet cooling
C*                coil.
C*******************************************************************
C*    INPUT VARIABLES
C*    MAir          Dry air mass flow rate                      (kg/s)
C*    TAirEnt       Entering air dry bulb temperature              (C)
C*    WAirEnt       Entering air humidity ratio                    (-)
C*    HAirEnt       Entering air enthalpy                       (J/kg)
C*    HAirLvg       Leaving air enthalpy                        (J/kg)
C*    UAExt         Heat transfer coefficient for external surface (W/C)
C*
C*    OUTPUT VARIABLES
C*    TAirLvg       Leaving air dry bulb temperature               (C)
C*    WAirLvg       Leaving air humidity ratio                     (-)
C*    Qsen          Sensible heat transfer rate                    (W)
C*    ErrStat       Error status indicator, 0 = ok, 1 = error      (-)
C*
C*    PROPERTIES
C*    Patm          Atmospheric pressure                           (-)
C*    CpAir         Specific heat of air                      (J/kg C)
C*    CpVap         Specific heat of water vapor              (J/kg C)
C*******************************************************************
C     MAJOR RESTRICTIONS:     Assumes condensate at uniform temperature.
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       TAIRSAT
C                             DRYBULB3
C                             HUMTH
C                             DEWPOINT
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              Elmahdy, A.H. and Mitalas, G.P.  1977.
C                             "A Simple Model for Cooling and
C                             Dehumidifying Coils for Use In Calculating
C                             Energy Requirements for Buildings,"
C                             ASHRAE Transactions,Vol.83 Part 2,
C                             pp. 103-117.
C*******************************************************************
C     INTERNAL VARIABLES:
C     capAir        Air capacity rate                            (W/C)
C     ntu           Number of heat transfer units                  (-)
C     effectiveness Heat exchanger effectiveness                   (-)
C     hCondSat      Saturated air enthalpy at temperature of
C                   condensate                                  (J/kg)
C     tempCond      Temperature of condensate                      (C)
C*******************************************************************
      REAL Prop(16)

      PARAMETER (Patm     = 1)
```

```
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      INTEGER ErrStat

      REAL ntu,MAir

      DATA small/1.E-9/

      ErrStat = 0

C1*** Determine the temperature effectiveness, assuming the temperature
C1*** of the condensate is constant (Cmin/Cmax = 0) and the specific heat
C1*** of moist air is constant
      capAir = MAir*(Prop(CpAir)+WAirEnt*Prop(CpVap))
      ntu = UAExt/MAX(capAir,small)
      effectiveness = 1 - EXP(-ntu)

C1*** Calculate coil surface enthalpy and temperature at the exit
C1*** of the wet part of the coil using the effectiveness relation
      effectiveness = MAX(effectiveness,small)
      hCondSat = HAirEnt-(HAirEnt-HAirLvg)/effectiveness

C1*** Calculate condensate temperature as the saturation temperature
C1*** at given saturation enthalpy
      tempCond = TAIRSAT(Prop,hCondSat)

C1*** Calculate exit air conditions and sensible heat transfer
      IF (tempCond .LT. DEWPOINT(Prop,WAirEnt)) THEN
        TAirLvg = TAirEnt-(TAirEnt-tempCond)*effectiveness
        WAirLvg = HUMTH(Prop,TAirLvg,HAirLvg)
      ELSE
        WAirLvg = WAirEnt
        TAirLvg = DRYBULB3(Prop,HAirLvg,WAirLvg)
      ENDIF

      Qsen = capAir*(TAirEnt-TAirLvg)

      RETURN
      END

      SUBROUTINE BYPASS(Prop,TEnt,WEnt,TLvg,WLvg,
     &                  TAdp,WAdp,BF,ErrStat)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C************************************************************************
C*    SUBROUTINE: BYPASS
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate apparatus dew point and bypass
C*                factor given entering and leaving moist
C*                air conditions of cooling coil.
```

```
C*********************************************************************
C*    INPUT VARIABLES
C*    TEnt          Entering air temperature                      (C)
C*    WEnt          Entering air humidity ratio                   (-)
C*    TLvg          Leaving air temperature                       (C)
C*    WLvg          Leaving air humidity ratio                    (-)
C*
C*    OUTPUT VARIABLES
C*    TAdp          Apparatus dewpoint temperature                (C)
C*    WAdp          Apparatus dewpoint humidity ratio             (-)
C*    BF            Bypass factor                                 (-)
C*    ErrStat       Error status indicator, 0 = ok, 1 = error     (-)
C*********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C                             Hugh Henderson & Kannan Rengarajan
C                             Florida Solar Energy Center
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       DEWPOINT
C                             ENTHALPY3
C                             HUMRATIO
C                             SATPRESS
C                             XITERATE
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*********************************************************************
C     INTERNAL VARIABLES:
C     hEnt          Entering air enthalpy
C     hLvg          Leaving air enthalpy
C     hAdp          Air enthalpy at apparatus dew point
C     slope         Ratio temperature difference to humidity difference
C                   between entering and leaving air states
C     tAdpEst       Estimate of TAdp from slope
C     error         Deviation of dependent variable in iteration
C     iter          Iteration counter
C     icvg          Iteration convergence flag
C     F1,F2         Previous values of dependent variable in XITERATE
C     X1,X2         Previous values of independent variable in XITERATE
C**********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,ERRSTAT

      REAL Prop(16)

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
```

```
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      DATA itmax/20/

C1*** Iterate to determine apparatus dewpoint at which the ADP
C1*** equals the temperature calculated by extending the line between
C1*** entering and leaving conditions to the saturation curve

C1*** Calculate "slope" of temperature vs. humidity ratio between
C1*** entering and leaving states
      slope = (TEnt-TLvg)/(WEnt-WLvg)

C1*** Initialize iteration parameters
      TAdp = DEWPOINT(Prop,WLvg)

      DO 100 iter=1,itmax
C1*** Calculate apparatus dewpoint and compare with predicted value
C1*** using entering conditions and slope
        WAdp = HUMRATIO(Prop(Patm),SATPRESS(Prop,TAdp))
        TAdpEst = TEnt - slope*(WEnt-WAdp)
        error = TAdp-TAdpEst
        TAdp = XITERATE(TAdp,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, exit loop
        IF (icvg .EQ. 1) GO TO 110
  100 CONTINUE

C1*** Apparatus dewpoint has not converged after maximum iterations.
C1*** Print error message, set return error flag, and RETURN
      WRITE(LUW,1001) itmax
 1001 FORMAT(/1X,'*** ERROR IN SUBOUTINE BYPASS ***'/
     &        1X,'    Apparatus dewpoint has not '
     &           'converged after ',I2,' iterations'/)
      ErrStat = 1

  110 CONTINUE

C1*** Calculate bypass factor from enthalpies
      hLvg = ENTHALPY3(Prop,TLvg,WLvg)
      hEnt = ENTHALPY3(Prop,TEnt,WEnt)
      hAdp = ENTHALPY3(Prop,TAdp,WAdp)
      BF = (hLvg-hAdp)/(hEnt-hAdp)

999   RETURN
      END

      REAL FUNCTION ENTHALPY3 (Prop,TDB,W)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: ENTHALPY3
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the enthalpy of moist air.
C***********************************************************************
C*    INPUT VARIABLES:
C*    TDB           Dry bulb temperature                           (C)
C*    W             Humidity ratio                                 (-)
C*
C*    OUTPUT VARIABLES:
```

```
C*    Enthalpy      Enthalpy of moist air                          (J/kg)
C*
C*    PROPERTIES:
C*    CpAir         Specific heat of air                         (J/kg C)
C*    CpVap         Specific heat of water vapor                 (J/kg C)
C*    Hfg           Reference heat of vaporization of water        (J/kg)
C***********************************************************************
C     MAJOR RESTRICTIONS      Uses perfect gas relationships
C                             Fit for enthalpy of saturated water vapor
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          PROP.INC
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

C1*** Calculate the enthalpy as a function of dry bulb temperature and
C1*** humidity ratio.
      hDryAir = Prop(CpAir)*TDB
      hSatVap = Prop(Hfg) + Prop(CpVap)*TDB
      Enthalpy3 = hDryAir + W*hSatVap

      RETURN
      END

      REAL FUNCTION DEWPOINT (Prop,W)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: DEWPOINT
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the dewpoint temperature given
C*              humidity ratio
```

```
C***********************************************************************
C*    INPUT VARIABLES
C*    W             Humidity ratio                                    (-)
C*
C*    OUTPUT VARIABLES
C*    DewPoint      Dew point temperature of air                      (C)
C*
C*    PROPERTIES
C*    Patm          Atmospheric pressure                             (Pa)
C***********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************
C     INTERNAL VARIABLES:
C     pw            Partial water vapor pressure                     (Pa)
C     small         Small number
C***********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      DATA small/1.E-9/

C1*** Test for "dry" air
      IF (W .LT. small) THEN
        DewPoint = -999
      ELSE
C1*** Calculate the partial water vapor pressure as a function of
C1*** humidity ratio.
        pw= Prop(Patm)*W/(.62198+W)
C1*** Calculate dewpoint as saturation temperature at water vapor
C1*** partial pressure
        DewPoint = SATTEMP(Prop,pw)
      ENDIF
```

```
  999 RETURN
      END

      REAL FUNCTION ENTHSAT (Prop,TDB)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*    FUNCTION: ENTHSAT
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the enthalpy at saturation
C*              for given dry bulb temperature
C*******************************************************************
C*    INPUT VARIABLES
C*    TDB           Dry bulb temperature                          (C)
C*
C*    OUTPUT VARIABLES
C*    EnthSat       Enthalpy at saturation                     (J/kg)
C*
C*    PROPERTIES
C*    Patm          Atmospheric pressure                         (Pa)
C*******************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          PROP.INC
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       SATPRESS
C                             HUMRATIO
C                             ENTHALPY3
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*******************************************************************
C     INTERNAL VARIABLES:
C     psat          Saturated water vapor pressure               (Pa)
C     w             Humidity ratio                                (-)
C*******************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
```

```
      PARAMETER (PAbsAdd  = 16)

C1*** Calculate the saturation pressure at the given temperature.
      psat = SATPRESS (Prop,TDB)

C1*** Calculate the humidity ratio from the saturation pressure
      w = HUMRATIO (Prop(Patm),psat)

C1*** Calculate the enthalpy as a function of dry bulb temperature
C1*** and humidity ratio.
      ENTHSAT = ENTHALPY3 (Prop,TDB,w)

      RETURN
      END

      REAL FUNCTION HUMRATIO (Patm,Pw)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: HUMRATIO
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the humidity ratio from water
C*              vapor pressure and atmospheric pressure
C***********************************************************************
C*    INPUT VARIABLES
C*    Patm          Atmospheric pressure                          (Pa)
C*    Pw            Partial water vapor pressure                  (Pa)
C*
C*    OUTPUT VARIABLES
C*    HumRatio      Humidity ratio                                 (-)
C***********************************************************************
C     MAJOR RESRICTIONS:      None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C************************************************************************

C1*** Calculate the humidity ratio.
      HumRatio = 0.62198*Pw/(Patm-Pw)

      RETURN
      END

      REAL FUNCTION HUMTH (Prop,TDB,H)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: HUMTH
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the humidity ratio of moist air
C*              from dry bulb temperature and enthalpy.
```

```
C***********************************************************************
C*    INPUT VARIABLES:
C*    H             Enthalpy                                       (J/kg)
C*    TDB           Dry bulb temperature                             (C)
C*
C*    OUTPUT VARIABLES:
C*    HumTH         Humidity ratio                                   (-)
C*
C*    PROPERTIES:
C*    CpAir         Specific heat of air                          (J/kg C)
C*    CpVap         Specific heat of water vapor                  (J/kg C)
C*    Hfg           Reference heat of vaporization of water         (J/kg)
C***********************************************************************
C     MAJOR RESTRICTIONS:     Uses perfect gas relationships
C                             Fit for enthalpy of saturated water vapor
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

C1*** Calculate humidity ratio from dry bulb temperature and enthalpy
C2*** hDryAir = Prop(CpAir)*TDB
C2*** hSatVap = Prop(Hfg) + Prop(CpVap)*TDB
C2*** Enthalpy = hDryAir + W*hSatVap
      HumTH = (H-Prop(CpAir)*TDB)/(Prop(Hfg)+Prop(CpVap)*TDB)

      RETURN
      END

      REAL FUNCTION RELHUM (Patm,Psat,HumRatio)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
```

```
C*    FUNCTION: RELHUM
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the relative humidity from
C*              saturation and atmospheric pressures
C***********************************************************************
C*    INPUT VARIABLES
C*    Patm          Atmospheric pressure                            (Pa)
C*    Psat          Saturation pressure                             (Pa)
C*    HumRatio      Humidity ratio                                   (-)
C*
C*    OUTPUT VARIABLES
C*    RelHum        Relative humidity                                (-)
C******      ****************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************
C     INTERNAL VARIABLES:
C     pw            Partial water vapor pressure                    (Pa)
C***********************************************************************

C1*** Calculate the partial water vapor pressure as a function of
C1*** humidity ratio.
      pw = Patm*HumRatio/(.62198+HumRatio)

C1*** Calculate the relative humidity as a function of partial water
C1*** vapor pressure and water vapor pressure at saturation.
      RelHum = pw/Psat

      RETURN
      END

      REAL FUNCTION RHOMOIST (RhoDry,W)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: RHOMOIST
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate moist air density from dry air
C*              density and humidity ratio
C***********************************************************************
C*    INPUT VARIABLES:
C*    RhoDry                  Dry air density                    (kg/m3)
C*    W                       Humidity ratio                         (-)
C*
C*    OUTPUT VARIABLES:
C*    RhoMoist                Density of dry air                 (kg/m3)
C***********************************************************************
C     MAJOR RESTRICTIONS:     None
C
```

```
C     DEVELOPER:               Shauna Gabel
C                              Michael J. Brandemuehl, PhD, PE
C                              University of Colorado at Boulder
C
C     DATE:                    January 1, 1992
C
C     INCLUDE FILES:           None
C     SUBROUTINES CALLED:      None
C     FUNCTIONS CALLED:        None
C
C     REVISION HISTORY:        None
C
C     REFERENCE:               1989 ASHRAE Handbook - Fundamentals
C*********************************************************************

C1*** Calculate the moist air density
      RhoMoist = RhoDry*(1.+W)

      RETURN
      END

      REAL FUNCTION SATPRESS (Prop,T)
C********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C********************************************************************
C*    SUBROUTINE: SATPRESS
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate saturation pressure of water
C*                vapor as a function of temperature
C********************************************************************
C*    INPUT VARIABLES
C*    T             Temperature                                    (C)
C*
C*    OUTPUT VARIABLES
C*    SatPress      Saturation pressure                           (Pa)
C*
C*    PROPERTIES
C*    TKelMult      Multiplying factor to convert user T to Kelvin
C*    TAbsAdd       Additive factor to convert user T to absolute T
C*                  tKel = Prop(TKelMult) * (T + Prop(TAbsAdd))
C*    PaMult        Multiplying factor to convert user P to Pascals
C*    PAbsAdd       Additive factor to convert user P to absolute P
C*                  Pa = Prop(PaMult) * (P + Prop(PAbsAdd))
C********************************************************************
C     MAJOR RESTRICTIONS:      173.16 K <= Temp <= 473.15 K
C
C     DEVELOPER:               Shauna Gabel
C                              Michael J. Brandemuehl, PhD, PE
C                              University of Colorado at Boulder
C
C     DATE:                    January 1, 1992
C
C     INCLUDE FILES:           prop.inc
C     SUBROUTINES CALLED:      None
C     FUNCTIONS CALLED:        None
C
C     REVISION HISTORY:        None
C
C     REFERENCE:               1989 ASHRAE Handbook - Fundamentals
C
C                              Hyland, R.W., and A. Wexler.  1983.
C                              Formulations for the thermodynamic
C                              properties of the saturated phases of H2O
C                              from 173.15 K to 473.15 K.  ASHRAE
```

```
C                              Transactions, Vol. 89, No. 2A, pp. 500-519
C***********************************************************************
C     INTERNAL VARIABLES:
C     tKel           Temperature in Kelvin                             (K)
C     pascals        Saturation pressure                              (Pa)
C***********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      DATA C1/-5674.5359/,C2/6.3925247/,C3/-0.9677843E-2/
      DATA C4/0.62215701E-6/,C5/0.20747825E-8/,C6/-0.9484024E-12/
      DATA C7/4.1635019/,C8/-5800.2206/,C9/1.3914993/,C10/-0.048640239/
      DATA C11/0.41764768E-4/,C12/-0.14452093E-7/,C13/6.5459673/

C1*** Convert temperature from user units to Kelvin.
      tKel = Prop(TKelMult)*(T+Prop(TAbsAdd))

C1*** If below freezing, calculate saturation pressure over ice.
      IF (tKel .LT. 273.15) THEN
        pascals = EXP(C1/tKel+C2+C3*tKel+C4*tKel**2+C5*tKel**3+C6*
     &                tKel**4+C7*ALOG(tKel))
C1*** If above freezing, calculate saturation pressure over liquid water.
      ELSE IF (tKel .GE. 273.15) THEN
        pascals = EXP(C8/tKel+C9+C10*tKel+C11*tKel**2+C12*tKel**3+C13
     &                *ALOG(tKel))
      ENDIF

C1*** Convert pressure from Pascals to user units
      SatPress = pascals/Prop(PaMult) - Prop(PAbsAdd)

      RETURN
      END

      REAL FUNCTION SATTEMP (Prop,P)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: SATTEMP
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the saturation (boiling)
C*              temperature of water given pressure
C***********************************************************************
C*    INPUT VARIABLES
C*    P              Pressure                                         (Pa)
```

```
C*
C*    OUTPUT VARIABLES
C*    SatTemp       Saturation temperature of water vapor            (C)
C*********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       SATPRESS
C                             XITERATE
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*********************************************************************
C     INTERNAL VARIABLES:
C     tSat          Water temperature guess                          (C)
C     pSat          Pressure corresponding to temp. guess           (Pa)
C     error         Deviation of dependent variable in iteration
C     iter          Iteration counter
C     icvg          Iteration convergence flag
C     F1,F2         Previous values of dependent variable in XITERATE
C     X1,X2         Previous values of independent variable in XITERATE
C*********************************************************************
CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA itmax/50/

C1*** Use an iterative process to determine the saturation temperature
C1*** at a given pressure using a correlation of saturated water vapor
C1*** pressure as a function of temperature

C1*** Initial guess of boiling temperature
      tSat = 100.

C1*** Iterate to find the saturation temperature
C1*** of water given the total pressure

C2*** Set iteration loop parameters
      DO 100 iter = 1,itmax
C1*** Calculate saturation pressure for estimated boiling temperature
        pSat = SATPRESS(Prop,tSat)
C1*** Compare with specified pressure and update estimate of temperature
        error = P - pSat
        tSat = XITERATE (tSat,error,X1,F1,X2,F2,iter,icvg)
C2*** If converged leave loop iteration
        IF (icvg .EQ. 1) GO TO 110
C2*** Water temperature not converged, repeat calculations with new
C2*** estimate of water temperature
  100 CONTINUE

C1*** Saturation temperature has not converged after maximum specified
C1*** iterations. Print error message, set return error flag, and RETURN
      WRITE(LUW,1001) itmax
 1001 FORMAT(/1X,'*** ERROR IN FUNCTION SatTemp ***'/
     &        1X,'    Saturation temperature has not '
     &           'converged after ',I2,' iterations'/)

  110 SatTemp = tSat
```

```
      RETURN
      END

      REAL FUNCTION TAIRSAT (Prop,HSat)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C***********************************************************************
C*    FUNCTION: ENTHSAT
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the dry bulb temperature given
C*              enthalpy at saturation.
C***********************************************************************
C*    INPUT VARIABLES:
C*    HSat          Enthalpy at saturation                         (J/kg)
C*
C*    OUTPUT VARIABLES:
C*    TAirSat       Dry bulb temperature                              (C)
C***********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       ENTHSAT
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C***********************************************************************
C     INTERNAL VARIABLES:
C     error         Deviation of dependent variable in iteration
C     iter          Iteration counter
C     icvg          Iteration convergence flag
C     F1,F2         Previous values of dependent variable in XITERATE
C     X1,X2         Previous values of independent variable in XITERATE
C***********************************************************************

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA itmax/20/,tSat/50./

C1*** Estimate saturation temperature if reasonable value not available
      IF(tSat .LT. -200. .OR. tSat .GT. 1000.) tSat = 50.

C1*** Calculate saturation temperature by iteration using function to
C1*** calculate saturation enthalpy from temperature
      DO 100 iter=1,itmax
        error = HSat - ENTHSAT(Prop,tSat)
        tSat = XITERATE(tSat,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave iteration loop.
        IF (icvg .EQ. 1) GO TO 110
C1*** Temperature not converged, repeat calculation with new
C1*** estimate of temperature.
  100 CONTINUE

C1*** Temperature has not converged after maximum specified
C1*** iterations. Print error message and RETURN
      WRITE(LUW,1001) itmax
```

```
 1001 FORMAT(/1X,'*** ERROR IN FUNCTION TAIRSAT ***'/
     &        1X,'    Temperature has not '
     &           'converged after ',I2,' iterations'/)

  110 CONTINUE

      TAirSat = tSat

      RETURN
      END


       REAL FUNCTION WETBULB (Prop,TDB,W)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*    FUNCTION: WETBULB
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate wet bulb temperature from dry
C*              bulb temperature and humidity ratio
C*******************************************************************
C*    INPUT VARIABLES
C*    TDB           Dry bulb temperature                         (C)
C*    W             Humidity ratio of air                        (-)
C*
C*    OUTPUT VARIABLES
C*    WetBulb       Wet bulb temperature                         (C)
C*
C*    PROPERTIES:
C*    Patm          Atmospheric pressure                        (Pa)
C*    Hfg           Latent heat of vaporization of water      (J/kg)
C*    CpAir         Specific heat of air                    (J/kg C)
C*    CpVap         Specific heat of water vapor            (J/kg C)
C*    CpWat         Specific heat of water                  (J/kg C)
C*******************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          prop.inc
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       SATPRESS
C                             HUMRATIO
C                             SATTEMP
C                             XITERATE
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*******************************************************************
C     INTERNAL VARIABLES:
C     tBoil         Boiling temperature of water at given pressure   (C)
C     psatStar      Saturation pressure at wet bulb temperature      (C)
C     wStar         Humidity  ratio as a function of PsatStar        (-)
C     newW          Humidity ratio calculated with wet bulb guess    (-)
C     error         Deviation of dependent variable in iteration
C     iter          Iteration counter
C     icvg          Iteration convergence flag
C     F1,F2         Previous values of dependent variable in XITERATE
C     X1,X2         Previous values of independent variable in XITERATE
```

```
C*********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      REAL newW

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA itmax/20/

C1*** Initial temperature guess
      tBoil = SATTEMP (Prop,Prop(Patm))
      WetBulb = MAX( MIN(WetBulb,TDB,(tBoil-0.1)), 0.)

C1*** Begin iteration loop
      DO 100 iter = 1,itmax
        IF (WetBulb .GE. (tBoil-0.09) ) WETBULB = tBoil-0.1
C1*** Determine the saturation pressure for wet bulb temperature
        psatStar = SATPRESS (Prop,WetBulb)
C1*** Determine humidity ratio for given saturation pressure
        wStar = HUMRATIO (Prop(Patm),psatStar)
C1*** Calculate new humidity ratio and determine difference from known
C1*** humidity ratio
        newW = ((Prop(Hfg)-(Prop(CpWat)-Prop(CpVap))*WetBulb)*wStar-
     &           Prop(CpAir)*(TDB-WetBulb))/(Prop(Hfg)+Prop(CpVap)*TDB
     &          -Prop(CpWat)*WetBulb)
C1*** Check error, if not satisfied, calculate new guess and iterate
        error = W-newW
        WetBulb = XITERATE(WetBulb,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave iteration loop.
        IF (icvg .EQ. 1) GO TO 900
C1*** Wet bulb temperature not converged, repeat calculation with new
C1*** estimate of wet bulb temperature.
  100 CONTINUE

C1*** Wet bulb temperature has not converged after maximum specified
C1*** iterations. Print error message, set return error flag, and RETURN
      WRITE(LUW,1009) itmax
 1009 FORMAT(/1X,'*** ERROR IN FUNCTION WetBulb ***'/
     &        1X,'    Wet bulb temperature has not '
     &           'converged after ',I2,' iterations'/)

  900 IF (WetBulb .GT. TDB) WetBulb = TDB

  999 RETURN
```

```
      END


       REAL FUNCTION DRYBULB3 (Prop,H,W)
C*****************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*****************************************************************
C*    FUNCTION: DRYBULB3
C*
C*    LANGUAGE: FORTRAN 77
C*
C*    PURPOSE:  Calculate the dry bulb temperature of
C*              moist air from enthalpy and humidity.
C*****************************************************************
C*    INPUT VARIABLES:
C*    H             Enthalpy                                 (J/kg)
C*    W             Humidity ratio                              (-)
C*
C*    OUTPUT VARIABLES:
C*    Drybulb3       Dry bulb temperature                        (C)
C*
C*    PROPERTIES:
C*    CpAir         Specific heat of air                   (J/kg C)
C*    CpVap         Specific heat of water vapor           (J/kg C)
C*    Hfg           Reference heat of vaporization of water  (J/kg)
C*****************************************************************
C     MAJOR RESTRICTIONS:     Uses perfect gas relationships
C                             Fit for enthalpy of saturated water vapor
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          PROP.INC
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              1989 ASHRAE Handbook - Fundamentals
C*****************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd

      REAL Prop(16)

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)
```

```
C1*** Calculate the dry bulb temperature as a function of enthalpy and
C1*** humidity ratio.
C2*** hDryAir = Prop(CpAir)*TDB
C2*** hSatVap = Prop(Hfg) + Prop(CpVap)*TDB
C2*** Enthalpy = hDryAir + W*hSatVap
      Drybulb3 = (H-Prop(Hfg)*W)/(Prop(CpAir)+Prop(CpVap)*W)

      RETURN
      END

      REAL FUNCTION XITERATE (X0,F0,X1,F1,X2,F2,ICount,ICvg)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*
C*    SUBROUTINE: XITERATE
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Iterately solves for the value of X which
C*                satisfies F(X)=0. Given Xi,F(Xi) pairs,
C*                the subroutine tests for convergence and
C*                provides a new guess for the value of the
C*                independent variable X.
C*******************************************************************
C*    INPUT VARIABLES
C*    F0            Current value of the function F(X)
C*    X0            Current value of X
C*    F1,F2         Two previous values of F(Xi)
C*    X1,X2         Two previous values of X
C*
C*        NOTE:     F1,X1,F2,X2 MUST BE STORED AND SAVED IN CALLING
C*                  ROUTINE.  THEY NEED NO INITIALIZATION
C*
C*    ICount        Number of iterations
C*
C*    OUTPUT VARIABLES
C*    XIterate      New estimate of X for F(X)=0
C*    ICvg          Convergence flag  ICvg = 0:  Not converged
C*                                    ICvg = 1:  Converged
C*******************************************************************
C     DEVELOPER:              Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              None
C*******************************************************************
C     INTERNAL VARIABLES
C     small         Small number used in place of zero
C     mode          Number of points used in fit
C                   mode = 1:  Use XPerburb to get new X
C                   mode = 2:  Linear equation to get new X
C                   mode > 2:  Quadratic equation to get new X
C     coef(i)       Coefficients for quadratic fit
C                   F(X) = coef(1) + coef(2)*X + coef(3)*X*X
C     check         Term under radical in quadratic solution
C     FiQ,XiQ       Double precision values of Fi,Xi
C     slope         Slope for linear fit
C     tolRel        Relative error tolerance
```

```
C     xPerturb      Perturbation applied to X to initialize iteration
C***********************************************************************
      DOUBLE PRECISION coef(3),check,F0Q,F1Q,F2Q,X0Q,X1Q,X2Q

      DATA tolRel/1.E-5/,xPerturb/0.1/,small/1.E-9/

C1*** Check for convergence by comparing change in X
      IF ((ABS(X0-X1) .LT. tolRel*MAX(ABS(X0),small) .AND.
     &     ICount .NE. 1) .OR. F0 .EQ. 0.) THEN
        XIterate = X0
        ICvg=1
        RETURN
      ENDIF

C1*** Not converged.
C2*** If after the second iteration there are enough previous points to
C2    fit a quadratic for the new X.  If the quadratic fit is not
C2    applicable, mode will be set to 1 or 2 and a new X will be
C2    determined by incrementing X from xPerturb or from a linear fit.
      ICvg=0
      mode=ICount
 10   IF (mode .EQ. 1) THEN
C1*** New guess is specified by xPerturb
        IF (ABS(X0) .GT. small) THEN
          XIterate = X0*(1.+xPerturb)
        ELSE
          XIterate = xPerturb
        ENDIF
      ELSEIF (mode .EQ. 2) THEN
C1*** New guess calculated from LINEAR FIT of most recent two points
        SLOPE=(F1-F0)/(X1-X0)
        IF(slope.EQ.0) THEN
          mode=1
          GO TO 10
        ENDIF
        XIterate=X0-F0/SLOPE
      ELSE
C1*** New guess calculated from QUADRATIC FIT
C1*** If two Xi are equal, set mode for linear fit and return to top
        IF (X0 .EQ. X1) THEN
          X1=X2
          F1=F2
          mode=2
          GO TO 10
        ELSEIF (X0 .EQ. X2) THEN
          mode=2
          GO TO 10
        ENDIF
C1*** Determine quadratic coefficients from the three data points
C1*** using double precision.
        F2Q=F2
        F1Q=F1
        F0Q=F0
        X2Q=X2
        X1Q=X1
        X0Q=X0
        coef(3)=((F2Q-F0Q)/(X2Q-X0Q)-(F1Q-F0Q)/(X1Q-X0Q))/(X2Q-X1Q)
        coef(2)=(F1Q-F0Q)/(X1Q-X0Q)-(X1Q+X0Q)*coef(3)
        coef(1)=F0-(coef(2)+coef(3)*X0Q)*X0Q
C1*** If points are colinear, set mode for linear fit and return to top
        IF (ABS(coef(3)) .LT. 1.D-10) THEN
          mode=2
          GO TO 10
        ENDIF
C1*** Check for precision.  If the coefficients do not accurately
C1*** predict the given data points due to round-off errors, set
```

```
C1*** mode for a linear fit and return to top.
        IF (ABS((coef(1)+(coef(2)+coef(3)*X1Q)*X1Q-F1Q)/F1Q) .GT.
     &     1.D-4) THEN
           mode=2
           GO TO 10
        ENDIF
C1*** Check for imaginary roots.  If no real roots, set mode to
C1*** estimate new X by simply incrementing by xPerturb
        check=coef(2)**2-4*coef(1)*coef(3)
        IF (check .LT. 0) THEN
C1*** Imaginary roots -- go back to linear fit
          mode=2
          GO TO 10
        ELSEIF (check .GT. 0) THEN
C1*** Real unequal roots -- determine root nearest to most recent guess
          XIterate=(-coef(2)+SQRT(check))/coef(3)/2
          xOther=-XIterate-coef(2)/coef(3)
          IF (ABS(XIterate-X0) .GT. ABS(xOther-X0)) XIterate=xOther
        ELSE
C1*** Real Equal Roots -- one solution
          XIterate=-coef(2)/coef(3)/2
        ENDIF
      ENDIF

C1*** Set previous variable values for the next iteration
      IF (mode .LT. 3) THEN
C1*** No valid previous points to eliminate.
        X2=X1
        F2=F1
        X1=X0
        F1=F0
      ELSE
C1*** Eliminate one previous point based on sign and magnitude of F(X)
C2*** Keep the current point and eliminate one of the previous ones.
        IF (F1*F0 .GT. 0 .AND. F2*F0 .GT. 0) THEN
C2*** All previous points of same sign.  Eliminate one with biggest F(X)
          IF (ABS(F2) .GT. ABS(F1)) THEN
            X2=X1
            F2=F1
          ENDIF
        ELSE
C1*** Points of different sign.
C1*** Eliminate the previous one with the same sign as current F(X).
          IF (F2*F0 .GT. 0)  THEN
            X2=X1
            F2=F1
          ENDIF
        ENDIF
        X1=X0
        F1=F0
      ENDIF

      RETURN
      END
```

Subroutine TYPE 77: Zone Return Air Mixing

SUBROUTINE AND FUNCTION CALL MAPPING

```
Return Air Temp - Calculate the humidity ratio of the return air stream from zone
latent loads
        SUBROUTINE TYPE77 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for
```

SOURCE CODE

```
      SUBROUTINE TYPE77 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*********************************************************************
C*    HVAC Thermal Distribution System Energy Calculations
C*********************************************************************
C*    SUBROUTINE: Return Air Temp
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the humidity ratio of
C*                the return air stream from zone
C*                latent loads
C*
C*********************************************************************
C*    INPUT VARIABLES DISCRIPTION(UNITS)                      SAMPLE VALUES
C*    XIN(1) MLvg   Dry air mass flow rate of primary air (kg/s) 10.0
C*    XIN(2) TLvg   Weighted Temperature of zone1 (C Kg)         30.0
C*    XIN(3) TLvg   Weighted Temperature of zone2 (C Kg)         30.0
C*    XIN(4) TLvg   Weighted Temperature of zone3 (C Kg)         30.0
C*    XIN(5) TLvg   Weighted Temperature of zone4 (C Kg)         30.0
C*    XIN(6) TLvg   Weighted Temperature of zone5 (C Kg)         30.0
C*    XIN(7) Wcc    Humidity ratio of primary air stream (-)       .0017
C*    XIN(8) Qlat   Total zones latent load  (W)              10000.
C*
C*    OUTPUT VARIABLES
C*    OUT(1) MAirRet Dry air mass flow rate of return   (kg/s)    3.65
C*    OUT(2) TAirRet    Temperature of zones (C)                 11.7877
C*    OUT(3) WAirRet    Humidity ratio of return air stream (-)    .00508950
C*    OUT(4) ErrStat Error flag (0=ok, 1=error) (-)               0.0
C*********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Ellen Franconi
C                             University of Colorado at Boulder
C
C     DATE:                   November 1997
C
C     INCLUDE FILES:          None
C     SUBROUTINES REQUIRED:   None
C     FUNCTIONS REQUIRED:     None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              None
C*********************************************************************
C     INTERNAL VARIABLES
C      none
C*********************************************************************
      DOUBLE PRECISION XIN, OUT

      INTEGER INFO,IOPT,NI,NP,ND

      REAL MLVG,MAirRet,CPAIR,CPVAP,HFG

      DIMENSION XIN(8),OUT(4),INFO(15)
```

```
      CHARACTER*3 YCHECK(8),OCHECK(4)

      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA small/1.E-9/, itmax/20/
      DATA CPAIR/1006.0/,CPVAP/1805.0/,HFG/250100.0/
      DATA YCHECK/'MF2','DM1','DM1','DM1','DM1','DM1','DM1','PW2'/
      DATA OCHECK/'MF2','TE1','DM1','DM1'/

      ErrStat = 0

      IOPT    = -1
      NI      = 8            !CORRECT NUMBER OF INPUTS
      NP      = 0            !CORRECT NUMBER OF PARAMETERS
      ND      = 0            !CORRECT NUMBER OF DERIVATIVES

      MLVG  = XIN(1)
      Wcc   = XIN(7)
      Qlat  = XIN(8)

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
C       CHECKS TO SEE IF USER'S INFO MATCHES CORRECT NUMBERS
        CALL RCHECK(INFO,YCHECK,OCHECK)
C       CHECKS TO SEE IF INPUT AND OUTPUT UNITS MATCH
        INFO(6)=4
      ENDIF

C1*** If Mair is zero, fan is off. Set values to zero and return
      IF (MLVG .EQ. 0.) THEN
        MAirRet=0.
        TAirRet=0.
        WAirRet=0.
        GO TO 999
      ENDIF

      TLvg  = (XIN(2)+XIN(3)+XIN(4)+XIN(5)+XIN(6))/XIN(1)
      WAirRet=Wcc+Qlat/(HFG*Mlvg)
      TAirRet=Tlvg
      MAirRet=Mlvg

 999  CONTINUE

      OUT(1)=MAirRet
      OUT(2)=TAirRet
      OUT(3)=WAirRet
      OUT(4)=ERRSTAT

      RETURN 1
      END
```

Subroutine TYPE 80: Economizer

```
SUBROUTINE AND FUNCTION CALL MAPPING
Model an outside air economizer controlled to mix outdoor and return air
C*                to a set mixed air temperature when
C*                the outside air conditions are beneficial
C*                for reducing cooling energy usage.
        SUBROUTINE TYPE80 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for

           CALL MIXOAIR (Prop,MAirAmb,TAirAmb,WAirAmb,MAirRet,TAirRet, -- subroutine
                                                                     S1 in type 80
     &                WAirRet,MAirMix,TAirMix,WAirMix,ErrStat)
           CALL MIXIAIR (Prop,MAirMix,TSetMix,TAirAmb,WAirAmb,TAirRet, -- subroutine
                                                                     S2 in type 80
     &                  WAirRet,MAirAmb,MAirRet,WAirMix,ErrStat)
           CALL MIXOAIR (Prop,MAirAmb,TAirAmb,WAirAmb,MAirRet,TAirRet, -- subroutine
                                                                     S1 in type 80
     &                WAirRet,MAirMix,TAirMix,WAirMix,ErrStat)
           hret = ENTHALPY3(Prop,TAirRet,WAirRet) -- function F2 in type 75
           hmix = ENTHALPY3(Prop,TAirMix,WAirMix) -- function F2 in type 75
           hoa = ENTHALPY3(Prop,TAirAmb,WAirAmb) -- function F2 in type 75

        S1 SUBROUTINE MIXOAIR (Prop,M1Ent,T1Ent,W1Ent,M2Ent,T2Ent,W2Ent,
     &                    MLvg,TLvg,WLvg,ErrStat)
              Calculate the leaving temperature, humidity ratio and mass flow rate of
              two mixed air streams by simple conservation.
           h1Ent = ENTHALPY3(Prop,T1Ent,W1Ent) -- function F2 in type 75
           h2Ent = ENTHALPY3(Prop,T2Ent,W2Ent) -- function F2 in type 75
           TLvg = DRYBULB3(Prop,hLvg,WLvg) -- function F13 in type 75

        S2 SUBROUTINE MIXIAIR (Prop,MLvg,TLvg,T1Ent,W1Ent,T2Ent,W2Ent,
     &                    M1Ent,M2Ent,WLvg,ErrStat)
              Calculate the mass flow rate of two entering air streams of a mixing box
              with a known leaving mass flow rate and the temperatures of all the
              streams.
           CALL MIXOAIR (PROP,M1Ent,T1Ent,W1Ent,M2Ent,T2Ent,W2Ent, -- subroutine S1
                                                                    in type 80
     &                mEst ,TLvg,WLvg,ErrStat)
           M1Ent = XITERATE(M1Ent,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                                in type 75

      1 PROP(PATM)     =  101325.0      Atmospheric pressure (Pa)
      2 PROP(CPAIR)    =    1006.0      Specific heat of dry air (J/kg C)
      3 PROP(CPWAT)    =    4186.0      Specific heat of liquid water (J/kg C)
      4 PROP(CPVAP)    =    1805.0      Specific heat of saturated water
                                        vapor (J/kg C)
      5 PROP(CPLIQ)    =    4186.0
      6 PROP(DVISCAIR) =        .0000182 Air dynamic viscosity (kg/m s)
      7 PROP(DVISCLIQ) =        .00144   Liquid dynamic viscosity (kg/m s)
      8 PROP(KAIR)     =        .026     Air thermal conductivity (W/m C)
      9 PROP(KLIQ)     =        .604     Liquid thermal conductivity (W/m C)
     10 PROP(RHOLIQ)   =     998.0      Liquid density (kg/m3)
     11 PROP(HFG)      = 2501000.0      Latent heat of vaporization of water (J/kg)
     12 PROP(RAIR)     =     287.055    Gas constant for air (J/kg C)
     13 PROP(TKELMULT) =       1.0      Multiplying factor to convert user
                                        T to Kelvin
     14 PROP(TABSADD)  =     273.15     Additive factor to convert user P to Kelvin:
                                        tKel = Prop(TKelMult)*T + Prop(TKelAdd)
     15 PROP(PAMULT)   =       1.0      Multiplying factor to convert user P to
                                        Pascals
     16 PROP(PABSADD)  =       0.0      Additive factor to convert user P to Pascals:
                                        Pa = Prop(PaMult)*P + Prop(PaAdd)
```

SOURCE CODE

```
      SUBROUTINE TYPE80 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C*******************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*******************************************************************
C*    SUBROUTINE: ECON
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    To model an outside air economizer
C*                controlled to mix outdoor and return air
C*                to a set mixed air temperature when
C*                the outside air conditions are beneficial
C*                for reducing cooling energy usage.
C*******************************************************************
C*    INPUT VARIABLES  DESCRIPTION(UNITS)                      SAMPLE VALUES
C*    XIN(1)  TAirRet  Return air dry bulb temperature(C)          23.8
C*    XIN(2)  WAirRet  Return air humidity ratio(-)                  .0077
C*    XIN(3)  TAirAmb  Outside air dry bulb temperature(C)         10.0
C*    XIN(4)  WAirAmb  Outside air humidity ratio(-)                 .0017
C*    XIN(5)  MAirMix  Mixed dry air mass flow rate(kg/s)           1.89
C*    XIN(6)  MAmbMin  Minimum outside air mass flow rate(kg/s)      .378
C*    XIN(7)  TSetMix  Mixed air temperature setpoint(C)           12.8
C*    XIN(8) VarClose  Ambient air control variable                10.0
C*    XIN(9) SetClose  Ambient air design parameter for            24.0
C*                     minimum damper position
C*    XIN(10)  HCMode  Heating or cooling mode indicator            1.0
C*                     Heating:  HCMode = 0
C*                     Cooling:  HCMode = 1
C*          Note:      Economizer cooling is considered to be unavailable
C*                     if VarClose > SetClose .OR. HCMode = 0
C*
C*    OUTPUT VARIABLES:
C*    OUT(1)  TAirMix  Mixed air temperature(C)                    12.8
C*    OUT(2)  WAirMix  Mixed air humidity ratio(C)                   .00290009
C*    OUT(3)  MAirRet  Return dry air mass flow rate(kg/s)           .378029
C*    OUT(4)  MAirAmb  Ambient dry air mass flow rate(kg/s)         1.510
C*    OUT(5)  QHe      Q added using econ  (W)                   -6000.
C*    OUT(6)  QCe      Q removed using economizer (W)             5000.
C*    OUT(7)  QHoa     Q added by outdoor air  (W)               -5000.
C*    OUT(8)  QCoa     Q removed by outdoor air (W)               1000.
C*******************************************************************
C     MAJOR ASSUMPTION:       None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     MIXOAIR
C                             MIXIAIR
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              ASHRAE.  1983.  Simplified Energy
C                             Analysis Using the Modified Bin Method,
C                             Atlanta: American Society of Heating,
C                             Refrigeration, and Air-Conditioning
C                             Engineers, Inc. pp.4-14-4-18.
C*******************************************************************
      REAL prop(16),MAMBMIN,MAIRMIX,MairAmb,MAirRet,QHe,QCe,QHoa,QCoa
      REAL hret,hmix,hoa
```

```
      DOUBLE PRECISION XIN, OUT

      INTEGER INFO(15),IOPT,NI,NP,ND

      DIMENSION XIN(10),OUT(8)

      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,errstat

      CHARACTER*3 YCHECK(10),OCHECK(8)

      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      PROP(PATM)     =  101325.0
      PROP(CPAIR)    =    1006.0
      PROP(CPWAT)    =    4186.0
      PROP(CPVAP)    =    1805.0
      PROP(CPLIQ)    =    4186.0
      PROP(DVISCAIR) =        .0000182
      PROP(DVISCLIQ) =        .00144
      PROP(KAIR)     =        .026
      PROP(KLIQ)     =        .604
      PROP(RHOLIQ)   =     998.0
      PROP(HFG)      = 2501000.0
      PROP(RAIR)     =     287.055
      PROP(TKELMULT) =       1.0
      PROP(TABSADD)  =     273.15
      PROP(PAMULT)   =       1.0
      PROP(PABSADD)  =       0.0

      DATA YCHECK/'TE1','DM1','TE1','DM1','MF2','MF2','TE1','TE1',
     &            'TE1','DM1'/
      DATA OCHECK/'TE1','DM1','MF2','MF2','PW2','PW2','PW2','PW2'/

      ErrStat = 0

      IOPT    = -1
      NI      = 10      !CORRECT NUMBER OF INPUTS
      NP      = 0       !CORRECT NUMBER OF PARAMETERS
      ND      = 0       !CORRECT NUMBER OF DERIVATIVES

      TAIRRET  = XIN(1)
      WAIRRET  = XIN(2)
      TAIRAMB  = XIN(3)
      WAIRAMB  = XIN(4)
      MAIRMIX  = XIN(5)
      MAMBMIN  = XIN(6)
```

```
      TSETMIX  = XIN(7)
      VARCLOSE = XIN(8)
      SETCLOSE = XIN(9)
      HCMODE   = XIN(10)

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
        CALL RCHECK(INFO,YCHECK,OCHECK)
        INFO(6)=8
      ENDIF

C1*** If Mair is zero, fan is off. Set flow to zero and return
      IF (MAIRMIX .EQ. 0.) THEN
        TAIRMIX=-99
        WAIRMIX=0
        MAIRRET=0
        MAIRAMB=0
        QHe=0
        QCe=0
        QHoa=0
        QCoa=0
        GO TO 999
      ENDIF

C1*** Determine whether economizer operation is available and whether
C1*** system is in heating or cooling mode

C2*** VarClose and SetClose are subroutine variables that allow a
C2    general comparison between two indices to evaluate the
C2    availability of economizer cooling.  The physical significance
C2    of VarClose and SetClose depend on the technique for economizer
C2    control.  Generally, economizer operation is available if
C2    VarClose < SetClose.  For example, for simple dry bulb temperature
C2    control of an economizer, VarClose would be the outdoor temperature
C2    and SetClose would be the setpoint for outdoor air temperature
C2    above which the economizer damper is set to minimum outdoor air.
C2    For more sophisticated enthalpy control, VarClose could be the
C2    outdoor enthalpy and SetClose could be the return air enthalpy,
C2    causing the economizer damper to close to minimum outdoor air if
C2    the outdoor air enthalpy is greater than the return air enthalpy.
      IF (VarClose .GT. SetClose .OR. NINT(HCMode) .EQ. 0) THEN
C1*** Economizer operation not available.  Close damper to minimum
C1*** and calculate mixed air conditions.
        MAirAmb = MAmbMin
        MAirRet = MAirMix-MAirAmb
        CALL MIXOAIR (Prop,MAirAmb,TAirAmb,WAirAmb,MAirRet,TAirRet,
     &                WAirRet,MAirMix,TAirMix,WAirMix,ErrStat)
      ELSE
C1*** Economizer operation available.  Calculate outdoor and return
C1*** airflow rates depending on comparison outdoor, return and mixed
C1*** air setpoint temperatures.
        IF ((TSetMix .GT. TAirAmb) .AND. (TSetMix .LT. TAirRet)) THEN
C1*** Normal economizer operation.
C1*** Calculate outdoor and return flow rates, ensuring that the outdoor
C1*** flow rate not less than minimum ventilation flow
          CALL MIXIAIR (Prop,MAirMix,TSetMix,TAirAmb,WAirAmb,TAirRet,
     &                  WAirRet,MAirAmb,MAirRet,WAirMix,ErrStat)
          IF (MAirAmb .LE. MAmbMin) THEN
            MAirAmb = MAmbMin
            MAirRet = MAirMix-MAirAmb
          ENDIF
        ELSEIF ((TAirRet-TSetMix) .GT. (TAirAmb-TSetMix)) THEN
C1*** Mixed air temperature setpoint is not between the return
C1*** and outdoor air temperatures and the ambient air temperature
C1*** less than the return air temperature
          MAirAmb = MAirMix
```

```
          MAirRet = 0.0
        ELSE
C1*** Mixed air temperature setpoint is not between the return
C1*** and outdoor air temperatures and the ambient air temperature
C1*** greater than the return air temperature
          MAirAmb = MAmbMin
          MAirRet = MAirMix-MAirAmb
        ENDIF
C1*** Calculate mixed air conditions for abnormal economizer operartion
        CALL MIXOAIR (Prop,MAirAmb,TAirAmb,WAirAmb,MAirRet,TAirRet,
     &               WAirRet,MAirMix,TAirMix,WAirMix,ErrStat)
      ENDIF

C1*** Calculate cooling contribution from economizer/outdoor air
C1*** negative => heating; positve => cooling
      hret = ENTHALPY3(Prop,TAirRet,WAirRet)
      hmix = ENTHALPY3(Prop,TAirMix,WAirMix)
      hoa = ENTHALPY3(Prop,TAirAmb,WAirAmb)

      Qecon=0.
      Qoa=0.
      QHe=0.
      QCe=0.
      QHoa=0.
      QCoa=0.

      IF (MAirAmb .GT. MAmbMin)  THEN
        Qecon = (hret-hmix)*MAIRAMB
      ELSE
        Qoa = (hoa-hret)*MAIRAMB
      ENDIF

      IF (Qecon .NE. 0) THEN
        IF (Qecon .LT. 0) THEN
          QHe = Qecon
        ELSE
          QCe = Qecon
        ENDIF
      ENDIF

      IF (Qoa .NE. 0) THEN
        IF (Qoa .LT. 0) THEN
          QHoa = Qoa
        ELSE
          QCoa = Qoa
        ENDIF
      ENDIF

999   Continue

      OUT(1) = TAIRMIX
      OUT(2) = WAIRMIX
      OUT(3) = MAIRRET
      OUT(4) = MAIRAMB
      OUT(5) = QHe
      OUT(6) = QCe
      OUT(7) = QHoa
      OUT(8) = QCoa

      RETURN 1
      END

      SUBROUTINE MIXOAIR (Prop,M1Ent,T1Ent,W1Ent,M2Ent,T2Ent,W2Ent,
     &                   MLvg,TLvg,WLvg,ErrStat)
C***********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
```

```
C*****************************************************************
C*    SUBROUTINE: MIXOAIR
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the leaving temperature,
C*                humidity ratio and mass flow rate of two
C*                mixed air streams by simple conservation.
C*****************************************************************
C*    INPUT VARIABLES
C*    M1Ent       Dry air mass flow rate of stream 1          (kg/s)
C*    T1Ent       Entering temperature of stream 1               (C)
C*    W1Ent       Entering humidity ratio of stream 1            (-)
C*    M2Ent       Dry air mass flow rate of stream 2          (kg/s)
C*    T2Ent       Entering temperature of stream 2               (C)
C*    W2Ent       Entering humidity ratio of stream 2            (-)
C*
C*    OUTPUT VARIABLES
C*    MLvg        Dry air mass flow rate of mixed stream      (kg/s)
C*    TLvg        Temperature of mixed stream                    (C)
C*    WLvg        Humidity ratio of mixed stream                 (C)
C*    ErrStat     Error flag (0=ok, 1=error)                     (-)
C*****************************************************************
C     MAJOR RESTRICTION:      None
C
C     DEVELOPER:              Shauna Gabel, MS
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     ENTHALPY3
C                             DRYBULB3
C     FUNCTIONS REQUIRED:     None
C
C     REVISION HISTORY:       None
C
C     REFERENCE:              None
C*****************************************************************
C*    INTERNAL VARIABLES
C*    small         Small number used in place of zero
C*****************************************************************
      REAL prop(16),M1Ent,M2Ent,MLvg,TLVG,WLVG

      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,errstat

      PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)
```

```
      DATA small/1.E-9/

      ErrStat = 0

C1*** Calculate the mass flow rate of the mixed stream.
      MLvg = M1Ent+M2Ent

C1*** If leaving flow is zero, set leaving conditions to those of
C1    stream 1 and RETURN.
      IF (ABS(MLvg) .LE. small) THEN
        WLvg = W1Ent
        TLvg = T1Ent
      ELSE
C1*** Leaving flow is not zero.  Proceed with calculations.
C1*** Calculate the humidity ratio of the mixed stream
        WLvg = (M1Ent*W1Ent+M2Ent*W2Ent)/MLvg
C1*** Calculate the mixed stream temperature from enthalpy and humidity
        h1Ent = ENTHALPY3(Prop,T1Ent,W1Ent)
        h2Ent = ENTHALPY3(Prop,T2Ent,W2Ent)
        hLvg = (M1Ent*h1Ent+M2Ent*h2Ent)/MLvg
        TLvg = DRYBULB3(Prop,hLvg,WLvg)
      ENDIF

      RETURN
      END
C***********************************************************************
C*    FILE: PROP.INC
C*
C*    This file assigns a numbers to air and water property names to be
C*    used in the "Prop" array.
C***********************************************************************
C     DEVELOPER:       Inger Andresen
C                      Michael J. Brandemuehl, PhD, PE
C
C     DATE:            July 1, 1991
C
C     FILES REQUIRED:  None
C***********************************************************************
C     INTERNAL VARIABLES:
C     Patm          Atmospheric pressure                            (Pa)
C     CpAir         Specific heat of dry air                    (J/kg C)
C     CpLiq         Specific heat of liquid water               (J/kg C)
C     CpVap         Specific heat of saturated water vapor      (J/kg C)
C     DViscAir      Air dynamic viscosity                       (kg/m s)
C     DViscLiq      Liquid dynamic viscosity                    (kg/m s)
C     KAir          Air thermal conductivity                     (W/m C)
C     KLiq          Liquid thermal conductivity                  (W/m C)
C     RhoLiq        Liquid density                               (kg/m3)
C     Hfg           Latent heat of vaporization of water          (J/kg)
C     RAir          Gas constant for air                        (J/kg C)
C     TKelMult      Multiplying factor to convert user T to Kelvin
C     TAbsAdd       Additive factor to convert user P to Kelvin
C                   tKel = Prop(TKelMult)*T + Prop(TKelAdd)
C     PaMult        Multiplying factor to convert user P to Pascals
C     PAbsAdd       Additive factor to convert user P to Pascals
C                   Pa = Prop(PaMult)*P + Prop(PaAdd)
C***********************************************************************
C      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
C     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
C     &        TKelMult,TAbsAdd,PaMult,PAbsAdd
C      REAL Prop(16)
C
C      PARAMETER (Patm     = 1)
C      PARAMETER (CpAir    = 2)
C      PARAMETER (CpWat    = 3)
C      PARAMETER (CpVap    = 4)
```

```
C      PARAMETER (CpLiq    = 5)
C      PARAMETER (DViscAir = 6)
C      PARAMETER (DViscLiq = 7)
C      PARAMETER (KAir     = 8)
C      PARAMETER (KLiq     = 9)
C      PARAMETER (RhoLiq   = 10)
C      PARAMETER (Hfg      = 11)
C      PARAMETER (RAir     = 12)
C      PARAMETER (TKelMult = 13)
C      PARAMETER (TAbsAdd  = 14)
C      PARAMETER (PaMult   = 15)
C      PARAMETER (PAbsAdd  = 16)
C
      SUBROUTINE MIXIAIR (Prop,MLvg,TLvg,T1Ent,W1Ent,T2Ent,W2Ent,
     &                    M1Ent,M2Ent,WLvg,ErrStat)
C*********************************************************************
C*    Copyright ASHRAE.  Toolkit for HVAC System Energy Calculations
C*********************************************************************
C*    SUBROUTINE: MIXIAIR
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculate the mass flow rate of two
C*                entering air streams of a mixing box with
C*                a known leaving mass flow rate and the
C*                temperatures of all the streams.
C*********************************************************************
C*    INPUT VARIABLES DISCRIPTION(UNITS)                  SAMPLE VALUES
C*    XIN(1) MLvg   Dry air mass flow rate of mixed stream (kg/s)
C*    XIN(2) TLvg   Temperature of mixed stream (C)
C*    XIN(3) T1En   Entering temperature of stream 1 (C)
C*    XIN(4) W1En   Entering humidity ratio of stream 1 (-)
C*    XIN(5) T2En   Entering temperature of stream 2 (C)
C*    XIN(6) W2Ent  Entering humidity ratio of stream 2 (-)
C*
C*    OUTPUT VARIABLES
C*    OUT(1) M1Ent   Dry air mass flow rate of stream 1 (kg/s)
C*    OUT(2) M2Ent   Dry air mass flow rate of stream 2 (kg/s)
C*    OUT(3) WLvg    Humidity ratio of mix air stream (-)
C*    OUT(4) ErrStat Error flag (0=ok, 1=error) (-)
C*********************************************************************
C     MAJOR RESTRICTIONS:     None
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          None
C     SUBROUTINES REQUIRED:   MIXOAIR
C     FUNCTIONS REQUIRED:     XITERATE
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              None
C*********************************************************************
C     INTERNAL VARIABLES
C     deltaT        Temperature difference of entering streams       (C)
C     mEst          Estimate of leaving flow                      (kg/s)
C     small         Small number, in place of zero
C     target        Target for mixed air temperatyure                (C)
C     error         Deviation of dependent variable in iteration
C     iter          Iteration counter
C     icvg          Iteration convergence flag
C     F1,F2         Previous values of dependent variable in XITERATE
```

```
C     X1,X2         Previous values of independent variable in XITERATE
C***********************************************************************
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir,
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,ERRSTAT

      REAL Prop(16),MLVG,M1ENT,M2ENT,WLVG,TLVG,T1ENT,W1ENT,
     &     T2ENT,W2ENT

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

       PARAMETER (Patm     = 1)
      PARAMETER (CpAir    = 2)
      PARAMETER (CpWat    = 3)
      PARAMETER (CpVap    = 4)
      PARAMETER (CpLiq    = 5)
      PARAMETER (DViscAir = 6)
      PARAMETER (DViscLiq = 7)
      PARAMETER (KAir     = 8)
      PARAMETER (KLiq     = 9)
      PARAMETER (RhoLiq   = 10)
      PARAMETER (Hfg      = 11)
      PARAMETER (RAir     = 12)
      PARAMETER (TKelMult = 13)
      PARAMETER (TAbsAdd  = 14)
      PARAMETER (PaMult   = 15)
      PARAMETER (PAbsAdd  = 16)

      DATA small/1.E-9/, itmax/20/

      ErrStat = 0

      deltaT = T2Ent-T1Ent

      IF(ABS(deltaT).LT.small) deltaT=small

C1*** Estimate the mass flow rate of stream 1 from a temperature balance
      M1Ent = (T2Ent-TLvg)/deltaT* MLvg
C1*** Set iteration loop parameters
      target = TLvg
C1*** BEGIN LOOP
      DO 100 iter = 1 ,itmax
C1*** Calculate leaving air temperature and humidity for estimated flows
        M2Ent = MLvg-M1Ent
        CALL MIXOAIR (PROP,M1Ent,T1Ent,W1Ent,M2Ent,T2Ent,W2Ent,
     &                mEst ,TLvg,WLvg,ErrStat)
C1*** Compare given leaving air temperature with estimated temperature
C1*** and determine new estimate of flow
        error = TLvg-target
        M1Ent = XITERATE(M1Ent,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave loop and RETURN
        IF (icvg .EQ. 1) GO TO 999
  100 CONTINUE

C1*** If not converged after itmax iterations, return error code

      WRITE(LUW,1005) itmax
 1005 FORMAT(/1X,'*** ERROR IN SUBROUTINE MIXIAIR ***'/
     &        1X,'    Temperature has not converged after, 'I2,
     &        '  iterations'/)
      ErrStat = 1

  999 RETURN
      END
```

Subroutine TYPE 81: Ceiling Return Plenum

SUBROUTINE AND FUNCTION CALL MAPPING

```
OPEN PLENUM RETURN - Return air temperature is calculated from a steady
C*                state energy balance on plenum. Heat gains/losses
C*                from lights, duct conduction, duct leakage,
C*                interior conduction, exterior conduction,
C*                and return air flow are included.
        SUBROUTINE TYPE81 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for
```

SOURCE CODE

```
      SUBROUTINE TYPE81 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C***********************************************************************
C*    SUBROUTINE: OPEN PLENUM RETURN
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Return air temperature is calculated from a steady
C*                state energy balance on plenum. Heat gains/losses
C*                from lights, duct conduction, duct leakage,
C*                interior conduction, exterior conduction,
C*                and return air flow are included.
C***********************************************************************
C*    INPUT VARIABLES DESCRIPTION(UNITS)                    SAMPLE VALUE
C*    XIN(1)   Mz     Zone air mass flow rate(kg/s)                 9.5
C*    XIN(2)   Tz     Zone air bulb temperature(C)                 22.5
C*    XIN(3)   HRz    Zone Humidity Ratio                            .009
C*    XIN(4)   Ta     Ambient Air Temperature      (C)              7.5
C*    XIN(5)   Qg     Internal gains from ceiling lights (W)     15000.0
C*    XIN(6)   Qc     Duct Conduction Heat Transfer (W)
C*    XIN(7)   Ql     Duct Leakage Heat Transfer (W)
C*    XIN(8)   Mlus   Mass leakage from duct upstream of boxes (kg/s) .05
C*    XIN(9)   HRl    Humidity Ratio of leakage air                  .004
C*    XIN(10)  Xlds   Downstream leakage fraction (-)
C*
C*    OUTPUT VARIABLES
C*    OUT(1)  Mp      Return Air Flowrate in Plenum (kg/s)         10.0
C*    OUT(2)  Tp      Return Air Temperature in Plenum (C)         25.5
C*    OUT(3)  HR      Return Air Humidity Ratio in Plenum            .008
C*    OUT(4) ErrStat  Error status indicator,0=ok,1=error(-)        0
C*
C*    PARAMETERS
C*    PAR(1)  UAe     Ext. perimeter heat transfer coef.  (J/C)
C*    PAR(2)  UAi     Int. ceil+floor heat transfer coef. (J/C)
C***********************************************************************
C     MAJOR RESTRICTIONS:     Model based on plenum located between
C                             two floors each having the same temp setpoint schedule
C
C     DEVELOPER:              Ellen Franconi
C                             Lawrence Berkeley Nat. Lab
C
C     DATE:                   February 9, 1998
C
C     INCLUDE FILES:          None
C     SUBROUTINES CALLED:     None
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:
C
C***********************************************************************
      DOUBLE PRECISION XIN, OUT
```

```
      DIMENSION XIN(10), OUT(4), PAR(2)
      DIMENSION INFO(15)

      INTEGER ErrStat, IOPT, NI, NP, ND, INFO

      REAL Mz,Mr,PAR,Mp,HRz,HRl,Mlus,Mlds

      CHARACTER*3 YCHECK(10), OCHECK(4)

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      DATA YCHECK/'MF2','TE1','DM1','TE1','PW2','PW2','PW2','MF2',
     & 'DM1','DM1'/
      DATA OCHECK/'MF2','TE1','DM1','DM1'/
      DATA PATM/101325.0/,CPAIR/1006.0/,CPVAP/1805.0/,HFG/2501000/,
     & RAIR/287.055/, TABSADD/273.15/

CPW20030417 Added next line to initialize ErrStat
      ErrStat = 0

       IOPT = -1.
      NI   = 10.     !CORRECT NUMBER OF INPUTS
      NP   = 2.      !CORRECT NUMBER OF PARAMETERS
      ND   = 0.      !CORRECT NUMBER OF DERIVATIVES

      Mz   = XIN(1)
      Tz   = XIN(2)
      HRz  = XIN(3)
      Ta   = XIN(4)
      Qg   = XIN(5)
      Qc   = XIN(6)
      Ql   = XIN(7)
      Mlus = XIN(8)
      HRl  = XIN(9)
      Xlds = XIN(10)

      UAe  = PAR(1)
      UAi  = PAR(2)

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
C      CHECKS #S IN USER SUPLLIED INFO ARRAY W/ NI, NP, AND ND
        CALL RCHECK(INFO,YCHECK,OCHECK)
C     CHECKS TO SEE IF THE UNITS ARE CONSISTENT
        INFO(6)=4
      ENDIF

C1*** If Mz is zero, fan is off. Set values and return
      IF (Mz .EQ. 0.) THEN
        Mp=0.
        Tp=-99.
        HRp=0.0001
        GO TO 999
      ENDIF

C1*** Calculate total UA from zones to plenum
      UApz=UAi+Mz*CPAIR

C1*** Solve for plenum/system return air temperature
      Tp=(Qg-Qc-Ql+(UAe*Ta)+(UApz*Tz))/(UAe+UApz)

C1*** Zone return + duct leakage = Plenum return
      Mlds=(Mz/(1-Xlds))-Mz
      Mp=Mz+Mlus+Mlds
```

```
      HRp=(HRz*Mz+HRl*(Mlus+Mlds))/Mp

999   Continue

      OUT(1) = Mp
      OUT(2) = Tp
      OUT(3) = HRp
      OUT(4) = ERRSTAT

      RETURN 1
      END
```

Subroutine TYPE 82: Upstream Ducts

SUBROUTINE AND FUNCTION CALL MAPPING

```
DUCT LOSS AND LEAKAGE- Calculates heat transfer from conduction and leakage
C*                in a ducted air stream.  Losses based on a log mean
C*                temperature difference between air stream and
C*                surroundings (i.e. plenum). Conduction loss determined
C*                analytically using the effectiveness/NTU method.
C                        Leakage rate is set at fixed CFM upstream of boxes and fixed
C                        % of flow downstream of boxes
        SUBROUTINE TYPE82 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
           CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
           CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for

           Talm = XITERATE(Talm,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                                in type 75
```

SOURCE CODE

```
      SUBROUTINE TYPE82 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
C************************************************************************
C*    Thermal Distribution System Model by EMFranconi
C************************************************************************
C*    SUBROUTINE: DUCT LOSS AND LEAKAGE
C*
C*    LANGUAGE:   FORTRAN 77
C*
C*    PURPOSE:    Calculates heat transfer from conduction and leakage
C*                in a ducted air stream.  Losses based on a log mean
C*                temperature difference between air stream and
C*                surroundings (i.e. plenum). Conduction loss determined
C*                analytically using the effectiveness/NTU method.
C************************************************************************
C*    INPUT VARIABLES   DESCRIPTION(UNITS)                    SAMPLE VALUE
C*    XIN(1)   Tai      Supply air temperature  (C)                  13.5
C*    XIN(2)   Mai      Fan air flowrate (kg/s)                      10.5
C*    XIN(3)   Mlus     Mass air leakage upstream of boxes (kg/s)      .10
C*    XIN(4)   Tpl      Temperature of air surrounding duct (C)      25.0
C*    XIN(5)   Xlds     Leakage fraction downstream of boxes (-)     25.0
C*    XIN(6)   TKGds    Box mass flow * downstream temperature (kg*C) 35.0
C*
C*    OUTPUT VARIABLES
C*    OUT(1)   Tboxes   Box inlet air temperature (C)                14.3
C*    OUT(2)   Mboxes   Box air flowrate (kg/s)                      10.0
C*    OUT(3)   Mzones   Zones supply air flowrate (kg/s)              9.0
C*    OUT(4)   Qcond    Conduction heat transfer  (W)              6000.
C*    OUT(5)   Qleak    Leakage heat transfer (W)                 10000.
C*    OUT(6)   ErrStat  Error status indicator,0=ok,1=error(-)        0
C*
C*    PARAMETERS
C*    PAR(1)  Ddct      Duct diameter (m)                              .30
C*    PAR(2)  Ldct      Duct length (m)                              20.
C*    PAR(3)  Ndct      Number of ducts (-)                          30
C*    PAR(4)  Rdct      Duct insulation R-value (C m2/W)              0.0
C*    PAR(5)  Losses    Loss/Leakage Calc Losses:
C                       0/off 1/leaks on 2/leaks+cond on (-)          1
C************************************************************************
C     MAJOR RESTRICTIONS: With a constant static pressure setpoint, system pressure
C                         changes with flowrate.  These changes may affect leakage
rate.
C                         This simple model does not attempt to quantify this effect.
C                         Leakage rate is set at fixed CFM upstream of boxes and fixed
C                         % of flow downstream of boxes
C
C     DEVELOPER:          Ellen Franconi
```

```
C                          Lawrence Berkeley Nat. Lab
C
C     DATE:                February 9, 1998
C
C     INCLUDE FILES:       None
C     SUBROUTINES CALLED:  None
C     FUNCTIONS CALLED:    None
C
C     REVISION HISTORY:    CPW20030418
C
C     REFERENCE:
C
C***********************************************************************
C     INTERNAL VARIABLES:
C     Talm     Log mean average air temp                    (C)
C     Taold    Last iterative value of Talm                 (C)
C     hco      Duct exterior convective coefficient         (W/m2 C)
C     hrad     Duct exterior radiation coefficient          (W/m2 C)
C     hout     Duct exterior effective heat transfer coef   (W/m2 C)
C     hin      Duct interior convection coefficienct        (W/m2 C)
C     htot     Overall duct heat transfer coefficient       (W/m2 C)
C     sv       Air specific volume                          (m3/kg)
C     v        Air velocity in duct                         (m/s)
C     emiss    Duct emissivity                              (-)
C     eff      effectiveness                                (-)
C     dTlm     Log mean temperature difference              (C)
C     small    Small number used in place of zero
C***********************************************************************
      DOUBLE PRECISION XIN, OUT

      DIMENSION XIN(6), OUT(6), PAR(5)
      DIMENSION INFO(15)

       INTEGER ErrStat, IOPT, NI, NP, ND, INFO, Losses

       REAL hco,hrad,hout,hin,htot,emiss,eff,Cmin,PAR,Mdct
      REAL Mai,Mboxes,Mzones,Mlus

       CHARACTER*3 YCHECK(6), OCHECK(6)

CPW20030418 Add next line to define I/O units
      COMMON /LUNITS/LUR,LUW,IFORM,LUK

       DATA YCHECK/'TE1','MF2','MF2','TE1','DM1','TE1'/
      DATA OCHECK/'TE1','MF2','MF2','PW2','PW2','DM1'/
      DATA PATM/101325.0/,CPAIR/1006.0/,pi /3.141592654/,
     & RAIR/287.055/, TABSADD/273.15/,SIGMA/.00000005669/
      DATA itmax/50/

CPW20030417 Added next line to initialize ErrStat
      ErrStat = 0

      IOPT = -1.
      NI      = 6. !CORRECT NUMBER OF INPUTS
      NP  = 5. !CORRECT NUMBER OF PARAMETERS
      ND  = 0. !CORRECT NUMBER OF DERIVATIVES

      Tai     = XIN(1)
      Mai     = XIN(2)
      Mlus    = XIN(3)
      Tpl     = XIN(4)
      Xlds    = XIN(5)
      TKGds   = XIN(6)

      Ddct    = PAR(1)
      Ldct    = PAR(2)
```

```
      Ndct    = PAR(3)
      Rdct    = PAR(4)
      Losses  = PAR(5)

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
C CHECKS #S IN USER SUPLLIED INFO ARRAY W/ NI, NP, AND ND
        CALL RCHECK(INFO,YCHECK,OCHECK)
C     CHECKS TO SEE IF THE UNITS ARE CONSISTENT
        INFO(6)=5
      ENDIF

      IF (Losses .EQ. 0 .OR. Mai .EQ. 0) THEN
        Qcond =0.
        Qleak =0.
        Tao =Tai
        Mboxes =Mai
        Mzones =Mai
      ELSEIF (Losses .EQ. 1) THEN
        Mboxes=Mai-Mlus
        Mzones=Mboxes*(1-Xlds)
        Tao=Tai
        Tlds=TKGds/Mboxes
        Qcond=0
        Qlus=Mlus*CPAIR*(Tpl-Tai)
        Qlds=Xlds*Mboxes*CPAIR*(Tpl-Tlds)
        Qleak=Qlus+Qlds
      ELSEIF (Losses .EQ. 2) THEN
        Mboxes=Mai-Mlus
        emiss=.80
        SV=.8131
C1*** Determine average duct flow rate based on 1/2 total losses
        Mdct=(Mboxes+.5*Mlus)/Ndct
        Talm=Tai
        Taold=Tai+.5   !First guess at log mean air temp
C1*** Forced convection,turbulent flow
        v=(Mdct*SV*4)/(pi*(Ddct**2))
        hin=8.80*((v**4)/Ddct)**(1/5)

        DO 100 iter = 1 ,itmax
C1*** Calculate heat transfer coefficients based on avg. temp
C2*** Free convection, turbulent flow
          hco=1.24*((Tpl-Talm)**(1/3))
          hrad=4*sigma*emiss*((((Talm+Tpl)/2)+273)**3)
          hout=1/((1/hco)+(1/hrad))
C2*** Overall heat transfer coefficient from duct air stream to surroundings
          Rtot=(1/hout)+Rdct+(1/hin)
          htot=1/Rtot
C1*** Conduction gain based on effectiveness-NTU method
C2*** Effectiveness based on m*CP air in plenum is infinite
          UA=htot*pi*Ddct*Ldct
          Cmin=Mdct*CPAIR
          eff=1-exp(-UA/Cmin)
          Qdc =eff*Cmin*(Tpl-Tai)
C1*** Calculate log mean temperature
          Taold=Talm
          Talm=Tpl-(Qdc/UA)
          error=Talm-Taold
          Talm = XITERATE(Talm,error,X1,F1,X2,F2,iter,icvg)
C1*** If converged, leave loop
          IF (icvg .EQ. 1) GO TO 999
  100   CONTINUE

C1*** If not converged after itmax iterations, return error code
        WRITE(LUW,105) itmax
  105   FORMAT(/1X,'*** ERROR IN SUBROUTINE DUCT_LEAK ***'/
```

```
     &          1X,'    Temperature has not converged after',I2,
     &          '  iterations'/)
        ErrStat = 1

 999    CONTINUE

        Qcond=Qdc*Ndct
        Qlus=Mlus*CPAIR*(Tpl-Talm)
        Qlds=Xlds*Mboxes*CPAIR*(Tpl-Tlds)
        Qleak=Qlus+Qlds
        Tao=Tai+(Qdc/Cmin)
      ENDIF

      OUT(1) = Tao
      OUT(2) = Mboxes
      OUT(3) = Mzones
      OUT(4) = Qcond
      OUT(5) = Qleak
      OUT(6) = ERRSTAT

      RETURN 1
      END
```

Subroutine TYPE 86: VAV Box and Downstream Ducts

SUBROUTINE AND FUNCTION CALL MAPPING

```
Zone Box w/ Downstream Leakage - Calculate the performance of a heating
C*                coil by modeling as a crossflow, both
C*                streams unmixed, heat exchanger. Results
C*                include outlet air temperature and
C*                humidity, outlet water temperature,
C*                sensible and total cooling capacities.
         SUBROUTINE TYPE86 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)
            CALL TYPECK(IOPT,INFO,NI,NP,ND) -- subroutine in TRNWIN\Kernal\typeck.for
            CALL RCHECK(INFO,YCHECK,OCHECK) -- subroutine in TRNWIN\Kernal\rcheck.for

            UATot = UAHX(capAir,TAirRat,capLiq,TLiqRat,QTotRat, -- function F1
                                                                 in type 75
     &             configHX,ErrStat)
            CALL DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt, -- subroutine S1
                                                                   in type 75
     &              UATot,configHX,
     &              TLiqLvg,TAirLvg,WAirLvg,QTot,ErrStat)
            mLiq = XITERATE(mLiq,error,X1,F1,X2,F2,iter,icvg) -- function F14
                                                               in type 75

      1 PROP(PATM)     =  101325.0       Atmospheric pressure (Pa)
      2 PROP(CPAIR)    =    1006.0       Specific heat of dry air (J/kg C)
      3 PROP(CPWAT)    =    4186.0       Specific heat of liquid water (J/kg C)
      4 PROP(CPVAP)    =    1805.0       Specific heat of saturated water
                                         vapor (J/kg C)
      5 PROP(CPLIQ)    =    4186.0
      6 PROP(DVISCAIR) =        .0000182 Air dynamic viscosity (kg/m s)
      7 PROP(DVISCLIQ) =        .00144   Liquid dynamic viscosity (kg/m s)
      8 PROP(KAIR)     =        .026     Air thermal conductivity (W/m C)
      9 PROP(KLIQ)     =        .604     Liquid thermal conductivity (W/m C)
     10 PROP(RHOLIQ)   =     998.0       Liquid density (kg/m3)
     11 PROP(HFG)      = 2501000.0       Latent heat of vaporization of water (J/kg)
     12 PROP(RAIR)     =     287.055     Gas constant for air (J/kg C)
     13 PROP(TKELMULT) =       1.0       Multiplying factor to convert user
                                         T to Kelvin
     14 PROP(TABSADD)  =     273.15      Additive factor to convert user P to Kelvin:
                                         tKel = Prop(TKelMult)*T + Prop(TKelAdd)
     15 PROP(PAMULT)   =       1.0       Multiplying factor to convert user P to
                                         Pascals
     16 PROP(PABSADD)  =       0.0       Additive factor to convert user P to Pascals:
                                         Pa = Prop(PaMult)*P + Prop(PaAdd)
```

SOURCE CODE

```
      SUBROUTINE TYPE86 (TIME,XIN,OUT,T,DTDT,PAR,INFO,ICNTRL,*)

C************************************************************************

C*     Thermal Distribution System Model by EMFranconi

C************************************************************************

C*    SUBROUTINE: Zone Box w/ Downstream Leakage

C*

C*    LANGUAGE:   FORTRAN 77

C*

C*    PURPOSE:    Calculate the performance of a heating
```

```
C*                 coil by modeling as a crossflow, both

C*                 streams unmixed, heat exchanger. Results

C*                 include outlet air temperature and

C*                 humidity, outlet water temperature,

C*                 sensible and total cooling capacities.

C*********************************************************************

C*    INPUT VARIABLES  DESCRIPTION(UNITS)                      SAMPLE VALUES

C*    XIN(1)     MLiq Liquid mass flow rate(kg/s)                     1.6

C*    XIN(2)  TLiqEnt Entering water temperature(C)                  62.2

C*    XIN(3)  MAirEnt Entering air mass flow rate (kg/s)              2.0

C*    XIN(4)  TAirEnt Entering air dry bulb temperature(C)           15.60
C*    XIN(5)  WAirEnt Entering air humidity ratio(-)                   .008

C*    XIN(6)    Tzone Zone temperature (C)                           25.0
C*    XIN(7)    QsenZ Zone sensible load (W)
C*    XIN(8)       lf Leakage fraction based on Mvav
C*

C*    OUTPUT VARIABLES

C*    OUT(1)  TLiqLvg Leaving water temperature(C)                   54.1365

C*    OUT(2)     MLiq Liquid flowrate (kg/s)                          0.8
C*    OUT(3)  TAirLvg Leaving air dry bulb temperature(C)            33.9

C*    OUT(4)  WAirLvg Leaving air humidity ratio(-)                    .008

C*    OUT(5)     MAir Box flowrate (kg/s)                             1.8
C*    OUT(6)    QSenC Sensible heat transfer rate(W)             -54005.8

C*    OUT(7)     TKGz Weighted zone air temperature                  45.0
C*    OUT(8)   LdsErr Zone load not met (0 or 1)                      0.0
C*    OUT(9)     Qcrh Reheat load when cooling                        0.0
C*    OUT(10)    Qhrh Reheat load when heating                        0.0
C*    OUT(11)    TKGb Weighted box air temperature                   45.0
C*    OUT(12) ErrStat Error status indicator,0=ok,1=error(-)          0.0
C*

C*    PARAMETERS

C*    PAR(1)  QTotRat Total heat transfer at rating(W)            44000.0

C*    PAR(2)  MLiqRat Liquid mass flow rate at rating(kg/s)           1.87

C*    PAR(3)  TLiqRat Entering water temperature at rating(C)        48.90

C*    PAR(4)  MAirRat Dry air mass flow rate at rating(kg/s)          6.8

C*    PAR(5)  TAirRat Entering air dry bulb temperature at rating(C) 15.6

C*    PAR(6)  WAirRat Entering air humidity ratio at rating(-)         .007

C*    PAR(7) Zone Box 1=CAV, 2=VAV                                    1
C*    PAR(8)      VAV turndown (%)                                   30.
C*
```

```
C*    PROPERTIES
C*    CpAir         Specific heat of dry air                    (J/kg C)
C*    CpLiq         Specific heat of liquid                     (J/kg C)
C*    CpVap         Specific heat of water vapor                (J/kg C)
C***********************************************************************
C     MAJOR RESTRICTIONS:     Models coil using effectiveness Ntu model
C                             as crossflow heat exchanger with both
C                             streams unmixed
C
C     DEVELOPER:              Shauna Gabel
C                             Michael J. Brandemuehl, PhD, PE
C                             University of Colorado at Boulder
C
C     DATE:                   January 1, 1992
C
C     INCLUDE FILES:          hcsim.inc
C                             prop.inc
C     SUBROUTINES CALLED:     DRYCOIL
C     FUNCTIONS CALLED:       None
C
C     REVISION HISTORY:       CPW20030418
C
C     REFERENCE:              None
C***********************************************************************
C     INTERNAL VARIABLES:
C     P(UATot)      Overall heat transfer coefficient              (W/C)
C     capAir        Air-side capacity rate                         (W/C)
C     capLiq        Water-side capacity rate                       (W/C)
C***********************************************************************
      DOUBLE PRECISION XIN, OUT
      INTEGER Patm,CpAir,CpWat,CpLiq,CpVap,DViscAir,
     &        DViscLiq,KAir,KLiq,RhoLiq,Hfg,RAir
     &        TKelMult,TAbsAdd,PaMult,PAbsAdd,iter,itmax
```

```
      INTEGER INFO,IOPT,NI,NP,ND,BOXTYPE,REHEAT

      REAL Prop(16),PAR,MLiqRat,MAirRat,MLiq,MAir,MinRate,
     &     LdsErr,MAirMin,TKGz,TKGb,Qcrh,Qhrh,lf

      DIMENSION XIN(8),OUT(12),PAR(8),INFO(15)

      CHARACTER*3 YCHECK(8),OCHECK(12)

CPW20030418 Add next line to define I/O units

      COMMON /LUNITS/LUR,LUW,IFORM,LUK

      PARAMETER (Patm     = 1)

      PARAMETER (CpAir    = 2)

      PARAMETER (CpWat    = 3)

      PARAMETER (CpVap    = 4)

      PARAMETER (CpLiq    = 5)

      PARAMETER (DViscAir = 6)

      PARAMETER (DViscLiq = 7)

      PARAMETER (KAir     = 8)

      PARAMETER (KLiq     = 9)

      PARAMETER (RhoLiq   = 10)

      PARAMETER (Hfg      = 11)

      PARAMETER (RAir     = 12)

      PARAMETER (TKelMult = 13)

      PARAMETER (TAbsAdd  = 14)

      PARAMETER (PaMult   = 15)

      PARAMETER (PAbsAdd  = 16)

      PROP(PATM)     =  101325.0

      PROP(CPAIR)    =    1006.0

      PROP(CPWAT)    =    4186.0

      PROP(CPVAP)    =    1805.0

      PROP(CPLIQ)    =    4186.0

      PROP(DVISCAIR) =        .0000182

      PROP(DVISCLIQ) =        .00144

      PROP(KAIR)     =        .026

      PROP(KLIQ)     =        .604

      PROP(RHOLIQ)   =     998.0

      PROP(HFG)      = 2501000.0
```

```
      PROP(RAIR)     =     287.055

      PROP(TKELMULT) =       1.0

      PROP(TABSADD)  =     273.15

      PROP(PAMULT)   =       1.0

      PROP(PABSADD)  =       0.0

      MLIQ    = XIN(1)

      TLIQENT = XIN(2)

      MAIR    = XIN(3)
      TAIRENT = XIN(4)

      WAIRENT = XIN(5)

      TZONE   = XIN(6)
      QSENZ   = XIN(7)
      LF      = XIN(8)

      QTOTRAT = PAR(1)

      MLIQRAT = PAR(2)

      TLIQRAT = PAR(3)

      MAIRRAT = PAR(4)

      TAIRRAT = PAR(5)

      WAIRRAT = PAR(6)

      BOXTYPE = PAR(7)

      MINRATE = PAR(8)

      DATA configHX/3./
      DATA YCHECK/'MF2','TE1','MF2','TE1','DM1','TE1','PW2','DM1'/
      DATA OCHECK/'TE1','MF2','TE1','DM1','MF2','PW2','DM1','DM1','PW2',
     &     'PW2','DM1','DM1'/

      DATA itmax/40/

      ErrStat = 0

      IOPT = -1

      NI  = 8  !CORRECT NUMBER OF INPUTS
      NP  = 8  !CORRECT NUMBER OF PARAMETERS
      ND  = 0  !CORRECT NUMBER OF DERIVATIVES

      IF (INFO(7).EQ.-1) THEN
        CALL TYPECK(IOPT,INFO,NI,NP,ND)
C  CHECKS TO SEE IF USER'S INFO MATCHES CORRECT NUMBER
        CALL RCHECK(INFO,YCHECK,OCHECK)
C  CHECKS TO SEE IF INPUT AND OUTPUT UNITS MATCH
        INFO(6)=11
      ENDIF

C1*** If QsenZ is zero, fan is off. Set flow to zero and return
      IF (Qsenz .EQ. 0.) THEN
        TLIQLVG=TLIQENT
```

```
        MLIQ=0.
        TAIRLVG=TAIRENT
        WAIRLVG=WAIRENT
        MAIR=0
        QSENC=0
        TKGz=0
        TKGb=0
        LdsErr=0
        Qcrh=0
        Qhrh=0
        GO TO 9999
C1*** If Qsenz is not zero but supply temp is zero - reset
      ELSEIF (TAIRENT .EQ. -99) THEN
        TAIRENT = 12.8
        WAIRENT =   .008
      ENDIF

C2*********************************************************************

C2    The code between these bars of asterisks is used to set internal

C2    parameters and is independent of component input values.  In an

C2    hourly simulation, this block of code may be skipped after the

C2    first call.

C1*** Calculate overall heat transfer coefficient from fluid states

C1*** and known total heat transfer

      capAir = MAirRat * (Prop(CpAir)+WAirRat*Prop(CpVap))

      capLiq = MLiqRat * Prop(CpLiq)

      UATot = UAHX(capAir,TAirRat,capLiq,TLiqRat,QTotRat,

     &             configHX,ErrStat)

C2*********************************************************************

C1*** Calculate box flowrate and delivery temp

      MAirMin= MAirRat
      REHEAT = 0

C2*** Calculations are based on air delivered to zone
C2*** Mzone = (1-lf)*Mair
      IF (BOXTYPE .EQ. 1) THEN
        Tzs=Tzone-(QsenZ/(MAir*(1-lf)*Prop(CpAir)))
        REHEAT=1
      ELSEIF (BOXTYPE .EQ. 2)  THEN
        MAir=-QsenZ/((1-lf)*Prop(CpAir)*(TAirEnt-Tzone))
        Tzs=TAirEnt
        IF (MAir < MAirMin)  THEN
          MAir=MAirMin
          Tzs=Tzone-(QsenZ/(MAir*(1-lf)*Prop(CpAir)))
          REHEAT=1
        ENDIF
      ENDIF

C3*** FLAG TO INDICATE COOL LOAD NOT MET
      IF (Tzs < TAirEnt) THEN
        Tzs = TAirEnt
        LdsErr=1
      ELSE
```

```
        LdsErr=0
      ENDIF

C1*** End zone flowrate and delivery temp calculations
C1*** CALCULATE WEIGHTED ZONE TEMP FOR DOWNSTREAM LEAKAGE Q TO PLENUM CALC
      TKGz=Mair*(1-lf)*Tzone

C1*** CALCULATE WEIGHTED ZONE TEMP FOR RETURN AIR CALC
      TKGb=Mair*Tzs

C1*** BEGIN COIL LOOP
      IF (REHEAT)  THEN
        DO 100 iter = 1 ,itmax
          CALL DRYCOIL (Prop,MLiq,TLiqEnt,MAir,TAirEnt,WAirEnt,

     &              UATot,configHX,

     &              TLiqLvg,TAirLvg,WAirLvg,QTot,ErrStat)

          QSenC = QTot

C1*** Compare given leaving air temperature with estimated temperature
C1*** and determine new estimate of flow
          error = TAirLvg-Tzs
          mLiq = XITERATE(mLiq,error,X1,F1,X2,F2,iter,icvg)

C1*** If converged, leave loop
          IF (icvg .EQ. 1) GO TO 999

C1*** If estimated flow is less than zero, set to small number
          IF(MLiq.LT.0) MLiq = 0.
  100   CONTINUE

      ELSE
        MLIQ=0.
        TLIQLVG=TLIQENT
        TAIRLVG=TAIRENT
        WAIRLVG=WAIRENT
        QSENC=0.
        GO TO 999
      ENDIF

C1*** If not converged after itmax iterations, return error code
      WRITE(LUW,1005) itmax
 1005 FORMAT(/1X,'*** ERROR IN SUBROUTINE ZONE_BOX ***'/
     &        1X,'    Temperature has not converged after',I2,
     &        '  iterations'/)
      ErrStat = 1


 999  CONTINUE

C1*** Tally Qcoil when zone requires heating or cooling
      IF (REHEAT) THEN
        IF(QsenZ .LT. 0) THEN
          Qhrh=QSenC
          Qcrh=0.
        ELSEIF (QsenZ .GT. 0) THEN
          Qhrh=0.
          Qcrh=QSenC
        ELSE
          Qhrh=0.
          Qcrh=0.
        ENDIF
      ELSE
        Qhrh=0.
```

```
          Qcrh=0.
        ENDIF

9999    CONTINUE

        OUT(1) = TLIQLVG
        OUT(2) = MLIQ

        OUT(3) = TAIRLVG

        OUT(4) = WAIRLVG
        OUT(5) = MAIR

        OUT(6) = QSENC
        OUT(7) = TKGz

        OUT(8) = LdsErr
        OUT(9) = Qcrh
        OUT(10) = Qhrh

        OUT(11) = TKGb
        OUT(12) = ERRSTAT


        RETURN 1

        END
```

California Energy Commission
Public Interest Energy Research Program

LBNL-53605
HPCBS # E4P22T1

HPCBS

High Performance Commercial Building Systems

Duct Thermal Performance Models

Element 4. Low Energy Cooling
Project 2.2 - Efficient Distribution Systems

Craig Wray
Lawrence Berkeley National Laboratory
October, 2003

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LBNL-53605

Duct Leakage Impacts on VAV System Performance in California Large Commercial Buildings

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October 2003

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TABLE OF CONTENTS

ABSTRACT

The purpose of this study is to evaluate the variability of duct leakage impacts on air distribution system performance for typical large commercial buildings in California. Specifically, a hybrid DOE-2/TRNSYS sequential simulation approach was used to model the energy use of a low-pressure terminal-reheat variable-air-volume (VAV) HVAC system with six duct leakage configurations (tight to leaky) in nine prototypical large office buildings (representing three construction eras in three California climates where these types of buildings are common). Combined fan power for the variable-speed-controlled supply and return fans at design conditions was assumed to be 0.8 W/cfm.

Based on our analyses of the 54 simulation cases, the increase in annual fan energy is estimated to be 40 to 50% for a system with a total leakage of 19% at design conditions compared to a tight system with 5% leakage. Annual cooling plant energy also increases by about 7 to 10%, but reheat energy decreases (about 3 to 10%). In combination, the increase in total annual HVAC site energy is 2 to 14%. The total HVAC site energy use includes supply and return fan electricity consumption, chiller and cooling tower electricity consumption, boiler electricity consumption, and boiler natural gas consumption.

Using year 2000 average commercial sector energy prices for California ($0.0986/kWh and $7.71/Million Btu), the energy increases result in 9 to 18% ($7,400 to $9,500) increases in HVAC system annual operating costs. Normalized by duct surface area, the increases in annual operating costs are 0.14 to 0.18 $/ft^2. Using a suggested one-time duct sealing cost of $0.20 per square foot of duct surface area, these results indicate that sealing leaky ducts in VAV systems has a simple payback period of about 1.3 years. Even with total leakage rates as low as 10%, duct sealing is still cost effective. This suggests that duct sealing should be considered at least for VAV systems with 10% or more total duct leakage.

The VAV system that we simulated had perfectly insulated ducts, and maintained constant static pressure in the ducts upstream of the VAV boxes and a constant supply air temperature at the air-handler. Further evaluations of duct leakage impacts should be carried out in the future after methodologies are developed to deal with duct surface heat transfer effects, to deal with airflows entering VAV boxes from ceiling return plenums (e.g., to model parallel fan-powered VAV boxes), and to deal with static pressure reset and supply air temperature reset strategies.

EXECUTIVE SUMMARY

Introduction. Despite the potential for significant energy savings by reducing duct leakage or other thermal losses from duct systems in large commercial buildings, California Title 24 has no provisions to credit energy efficient duct systems in these buildings. A substantial reason is the lack of readily available simulation tools to demonstrate the energy saving benefits associated with efficient duct systems in large commercial buildings. A related reason is that, although substantial energy increases due to duct leakage have been identified by recent field work and simulations, the variability of these impacts for the different building vintages and climates in California has not been established.

Purpose. The overall goal of the Efficient Distribution Systems (EDS) project within the PIER High Performance Commercial Building Systems Program is to bridge the gaps in current duct thermal performance modeling capabilities, and to expand our understanding of duct thermal performance in California large commercial buildings. As steps toward this goal, our strategy in the EDS project involves two parts: 1) developing a whole-building energy simulation approach for analyzing duct thermal performance in large commercial buildings, and 2) using the tool to identify the energy impacts of duct leakage in California large commercial buildings, in support of future recommendations to address duct performance in the Title 24 Energy Efficiency Standards for Nonresidential Buildings.

Objectives. The specific technical objectives for the EDS project were to:

1. Identify a near-term whole-building energy simulation approach that can be used in the impacts analysis task of this project (see Objective 3), with little or no modification. A secondary objective is to recommend how to proceed with long-term development of an improved compliance tool for Title 24 that addresses duct thermal performance.
2. Develop an Alternative Calculation Method (ACM) change proposal to include a new metric for thermal distribution system efficiency in the reporting requirements for the 2005 Title 24 Standards. The metric will facilitate future comparisons of different system types using a common "yardstick".
3. Using the selected near-term simulation approach, assess the impacts of duct system improvements in California large commercial buildings, over a range of building vintages and climates. This assessment will provide a solid foundation for future efforts that address the energy efficiency of large commercial duct systems in Title 24.

This report presents findings and recommendations that resulted from our modeling efforts related to duct thermal performance (Objective 3).

Outcomes. There are two principal outcomes from the work reported here:

<u>Uniformity of Duct Leakage Impacts</u>: A hybrid DOE-2/TRNSYS sequential simulation approach was used to model the energy use of a low-pressure terminal-reheat variable-air-volume HVAC system with six duct leakage configurations (tight to leaky) in nine prototypical large office buildings (representing 1980s, 1990s, and 2005 construction eras in three California climates where these types of buildings are common – Oakland, Pasadena, and Sacramento). Combined fan power for the variable-speed-controlled supply and return fans at design conditions was assumed to be 0.8 W/cfm.

Based on our analyses of the 54 simulation cases, we conclude that there can be substantial energy impacts due to duct leakage in this type of building. This finding is consistent with recent field measurements in a large office building in Sacramento. Our analyses indicate that a leaky VAV system (19% total duct leakage) will use about 40 to 50% more fan energy annually than a tight system with 5% leakage. Annual cooling plant energy also increases by about 7 to 10%, but reheat energy decreases (about 3 to 10%). In combination, the increase in total annual HVAC site energy is 2 to 14%. The total HVAC site energy use includes supply and return fan electricity consumption, chiller and cooling tower electricity consumption, boiler electricity consumption, and boiler natural gas consumption.

Using year 2000 average commercial sector energy prices for California ($0.0986/kWh and $7.71/Million Btu), the energy increases result in 9 to 18% ($7,400 to $9,500) increases in HVAC system annual operating costs. Our simulations also indicate that climate and building vintage differences do not cause significant variability in duct leakage impacts on fan energy use or on operating cost for leaky duct systems. This suggests that a single duct leakage threshold could be developed for use in the Title 24 prescriptive compliance approach and would not need to be climate or building age specific.

Duct Sealing is Cost Effective: Normalized by duct surface area, the increases in HVAC system annual operating costs are 0.14 to 0.18 $\$/ft^2$ for the 19% leakage case. Using a suggested one-time duct sealing cost of $\$0.20/ft^2$ of duct surface area, these results indicate that sealing leaky ducts in VAV systems has a simple payback period of about 1.3 years. Even with total leakage rates as low as 10%, duct sealing is still cost effective. This suggests that duct sealing should be considered at least for VAV systems with 10% or more total duct leakage.

Recommendations. Before duct performance in large commercial buildings can be accounted for in Title 24 nonresidential building energy standards, there are several issues that must be addressed and resolved. These include:

1. Specifying reliable duct air leakage measurement techniques that can be practically applied in the large commercial building sector.
2. Defining the duct leakage condition for the standard building used in Title 24 compliance simulations.
3. Assuring consistency between simulated duct performance impacts and actual impacts.
4. Developing compliance tests for the Alternative Calculation Method (ACM) Approval Manual (CEC 2001b) to evaluate duct performance simulations.

Three additional steps will be required to further develop duct-modeling capabilities that address limitations in existing models and to initiate strong market activity related to duct system improvements. We recommend that these steps include:

1. Implementing duct models in user-friendly commercially-available software for building energy simulation, validating the implementations with case studies and demonstrations, and obtaining certification for software use as a primary or alternative compliance tool in support of the Title 24 Nonresidential Standards.
2. Developing methodologies to deal with airflows entering VAV boxes from ceiling return plenums (e.g., to model parallel fan-powered VAV boxes), to deal with duct surface heat transfer effects, and to deal with static pressure reset and supply air temperature reset strategies.
3. Transferring information to practitioners through publications, conferences, workshops, and other education programs.

1. INTRODUCTION

1.1 Background

Previous research suggests that duct systems in California commercial buildings suffer from a number of problems, such as thermal losses due to duct air leakage. For example, measurements by Diamond et al. (2003) in a large commercial building confirmed predictions by Franconi et al. (1998) that duct leakage can significantly increase HVAC system energy consumption: adding 15% duct leakage at operating conditions leads to a fan power increase of 25 to 35%. Diamond et al. also estimated that eliminating duct leakage airflows in half of California's existing large commercial buildings has the potential to save about 560 to 1,100 GWh annually ($60-$110 million per year or the equivalent consumption of 83,000 to 170,000 typical California houses), and about 100 to 200 MW in peak demand.

California Title 24, Part 6 (CEC 2001a) is one of the most advanced energy codes in the United States. The impacts of duct thermal performance in residences are already addressed by Title 24 compliance procedures; duct-system energy efficiency requirements have recently been added for small commercial buildings with individual packaged equipment serving 5,000 ft² or less where ducts are located in spaces between insulated ceilings and the roof, or outside the building; and new requirements for duct performance in other small commercial buildings are being developed. However, despite the potential for significant energy savings by reducing thermal losses from duct systems in large commercial buildings, Title 24 has no provisions to credit energy efficient duct systems in these buildings. A substantial reason is the lack of readily available simulation tools to demonstrate the energy saving benefits associated with efficient duct systems in large commercial buildings. A related reason is that, although substantial energy increases due to duct leakage have been identified, the variability of these impacts for the different building vintages and climates in California has not been established.

1.2 Project Objectives

The work reported here is part of the Efficient Distribution Systems (EDS) project within the PIER High Performance Commercial Building Systems Program. The EDS project goal is to bridge the gaps in duct system modeling capabilities, and to expand our understanding of duct thermal performance in California's large commercial buildings, by following through on the strategy outlined by Xu et al. (1999). As steps toward this goal, the project involves three specific technical objectives:

1. Identify a near-term whole-building energy simulation approach that can be used in the impacts analysis task of this project (see Objective 3), with little or no modification. A secondary objective is to recommend how to proceed with long-term development of an improved compliance tool for Title 24 that addresses duct thermal performance.
2. Develop an Alternative Calculation Method (ACM) change proposal to include a new metric for thermal distribution system efficiency in the reporting requirements for the 2005 Title 24 Standards. The metric will facilitate future comparisons of different system types using a common "yardstick".
3. Using the selected near-term simulation approach, assess the impacts of duct system improvements in California large commercial buildings, over a range of building vintages and climates. This assessment will provide a solid foundation for future efforts that address the energy efficiency of large commercial duct systems in Title 24.

In support of Objective 1, Wray (2003) carried out a review of documents related to past HVAC system modeling efforts, which was supplemented by discussions with other simulation experts. Based on that work, he defined a set of modeling principles and published HVAC component models that can be used to guide duct thermal performance modeling for large commercial buildings. He also suggested that the best short-term approach for evaluating duct leakage impacts on HVAC system performance is to build upon past research that used DOE-2 and TRNSYS sequentially (Franconi 1999).

However, Wray (2003) concluded that DOE-2 is not a suitable platform for the long-term development of models to address duct system performance in large commercial buildings. He suggested instead that EnergyPlus, which is based in part on DOE-2, be developed to include component models like the TRNSYS ones identified for use in this project's duct leakage impact analysis task. Although EnergyPlus has no duct performance models, we expect that the recommended enhancements could be applied in a relatively straightforward manner.

Regarding Objective 2, the California Energy Commission has accepted the ACM change that Modera (2002) proposed for the 2005 Title 24 Standards to address HVAC distribution system efficiency in large commercial buildings. The metric of interest, HVAC Transport Efficiency, characterizes the overall efficiency of the thermal distribution system as the ratio between the energy expended to transport heating, cooling, and ventilation throughout a building and the total thermal energy delivered to the various conditioned zones in the building. Since the proposal is for a set of reporting changes, the ACM proposal should not require significant effort on the part of ACM providers to implement the changes in existing Title 24 non-residential compliance software.

Objective 3 is the focus of the work reported here. In particular, this report presents findings and recommendations that resulted from our modeling efforts to assess the impacts of duct thermal performance improvements.

This project contributes to the PIER program objective of improving the energy cost and value of California's electricity in two ways. One is by developing analytical methods to show that well designed duct systems in large commercial buildings can save much of the energy used to move and condition air. The other is by making progress toward new requirements for commercial duct system efficiency in future revisions of Title 24. We expect that the new analytical capabilities and performance requirements will ultimately result in smaller capacity, more energy-efficient building systems, which will also reduce peak electrical demand from California's commercial building sector and improve the reliability and quality of California's electricity.

1.3 Report Organization

In **Section 2, California Duct Systems**, we briefly describe duct system types that are common in California large commercial buildings, and present an example to illustrate the effects of duct system deficiencies.

In **Section 3, Modeling Approach**, we summarize the DOE-2/TRNSYS simulation approach that we used to evaluate the impacts of duct leakage on VAV system performance.

In **Section 4, Building and HVAC System Characteristics**, we describe the characteristics of the prototypical large office building that we simulated, and summarize the 54 building vintage, climate, and duct leakage combinations that we used in this study.

In **Section 5, System - Plant Energy Use Regressions**, we summarize our approach to translate TRNSYS air-handling system coil loads into cooling and heating plant energy use.

In **Section 6, Results**, we describe the impacts of duct leakage on building energy performance, based on the simulation results. To improve readability, the large data tables referred to in this section are located after the **References** section.

In **Section 7, Conclusions**, we present what we learned from the research.

In **Section 8, Other Issues and Implications**, we recommend future activities.

Following the **Glossary** and **References**, there are two **Appendices**:

"**Appendix I, Building Schedules**" lists the various operating schedules that we used in the simulations.

"**Appendix II, Regression Equations and Coefficients**" provides details about the system - plant energy use regressions that we developed, and explains how they are used.

2. CALIFORNIA DUCT SYSTEMS

The information in this section briefly describes duct system types that are common in California large commercial buildings, and presents an example to illustrate the effects of duct system deficiencies. The intent of this section is to help the reader understand why we simulated VAV systems in large office buildings and to conceptualize how duct leakage can affect the performance of an HVAC system.

2.1 Duct System Types

Using survey data collected from 1988 through 1993 by or for California utilities and for the California Energy Commission, Modera et al. (1999) determined that there are three basic types of duct systems in California commercial buildings:

- *Single-duct* systems generate either a cool or warm air stream at the air-handler. The supply air is delivered to the conditioned zones through a single duct system connected to the air-handler. Reheat coils at individual terminal units can be used to add heat to the supply air when needed.
- *Dual duct* systems generate a cool air stream and a warm air stream at the air-handler. Each air stream is supplied to terminal boxes through a separate duct system. The terminal boxes mix the air streams before the supply air enters the zones.
- *Multizone* duct systems also generate a cool air stream and a warm air stream at the air-handler, but they use dampers at the air-handler instead of at a terminal box to mix the cool and warm air streams for each zone. Each zone's supply air is delivered through a separate duct system (this system is somewhat like several single-duct systems operating in parallel).

All of these duct systems use one of two methods to control the amount of energy supplied to each zone. A *constant-air-volume (CAV)* system delivers a fixed quantity of supply air to the conditioned space and maintains desired conditions by varying the temperature of the supply air. A *variable-air-volume (VAV)* system maintains space temperature by varying the quantity of supply air, generally at a fixed temperature.

Based on the floor area served by these duct systems (Modera et al. 1999), the most common system across different California building types is the single duct CAV system (71%). The next most common system type is the multizone system (19%). Single-duct VAV systems (8%) and dual duct systems (2%) serve the remainder of the floor area. Note that the fraction of multizone systems might be overrepresented by these data. Modera et al. indicated that the survey data may include some inappropriate affirmative responses for multizone systems. In some cases, the respondent may have called a system that serves more than one zone a multizone system, even though the system is not really a multizone system as described above. For example, some of the multizone systems might actually be single-duct VAV systems that serve multiple zones.

The fractions of floor areas served by CAV and VAV system types are difficult to determine, because the fractions for multizone and dual-duct systems are unknown. However, based on data from Modera et al. (1999) and EIA (2002), the fraction of VAV systems may be in the range of 8 to 34% of the total building floor area. The EIA data indicate that VAV systems serve 34% of the large commercial building floor area in the U.S. Pacific region, which includes California.

Although there are substantially fewer VAV systems than CAV systems in California, it is clear that VAV systems are used in a significant fraction of California buildings and need to be addressed when developing duct models for large commercial buildings. A reason to focus on VAV systems is that if one is able to model a VAV system, then a CAV system can also be modeled (it is a simplification of a VAV system). Another reason is that an EPRI study (Pietsch 1991) suggested a significant national trend over the past 30 years towards the use of VAV systems in new construction (e.g., about 75% of new duct systems in the period 1980 through 1990 were VAV systems).

Of the floor area served by single-duct VAV systems, the data from Modera et al. (1999) indicate that most (98%) of it is in large office buildings; the remainder (2%) is primarily in hotel and retail buildings. For this reason, we focused on large office buildings in our study.

2.2 Effects of Duct Deficiencies

In large commercial buildings, duct systems and the effects of deficiencies in these systems are much more complex than in most residential and small-commercial buildings. As an example to illustrate the effects of duct system deficiencies, consider a large commercial building equipped with a single-duct terminal-reheat VAV system that has leaky supply ducts located within a ceiling return air plenum.

When conditioned air leaks from the supply ducts, the heating or cooling energy associated with leakage heats or cools the return air and changes its temperature (and enthalpy). Depending on the temperature difference across each surface that separates the plenum from adjacent conditioned spaces and the outdoors, some of the energy associated with the leakage airflow is transferred from the plenum by conduction across these surfaces. The energy transferred by conduction between the plenum and adjacent zones may be beneficial or detrimental to zone loads. For example, when there is simultaneous heating of perimeter zones and cooling of the core zone, the heating energy associated with leakage from ducts that serve the perimeter zones will tend to increase plenum temperatures; the cooling energy associated with leakage from ducts that serve the core zone will tend to decrease plenum temperatures. A net increase in plenum temperatures will increase the core-zone cooling load and decrease the perimeter-zone heating loads. Conversely, a net decrease in plenum temperatures will decrease the core-zone cooling load and increase the perimeter-zone heating loads.

If the VAV boxes deliberately induce airflows from the ceiling plenum (driven by induction effects or by VAV box fans), the change in return air enthalpy affects the mixed supply air enthalpy within and downstream of the VAV box. This in turn affects the energy that is transferred to the conditioned spaces by these airflows. It can also affect VAV box reheat coil loads (e.g., reduced return air enthalpy due to cool supply air leakage upstream of the VAV box or from other ducts reduces the VAV box mixed air enthalpy and increases reheat coil loads).

A change in return air temperature due to duct leakage will also change cooling coil loads when the economizer is not operating. For example, consider an air-handler with an economizer that is controlled based on dry-bulb temperatures (rather than on enthalpies). When the outdoor air temperature is above the return air temperature high-limit set point, the amount of outdoor air entering the air-handler is the minimum required for ventilation. The remainder of the mixed airflow entering the air-handler (same flow rate as the supply airflow) is return air. Mechanical cooling is used to maintain the desired supply air temperature. In this case, the change in return air enthalpy due to duct leakage will affect the mixed air enthalpy entering the air-handler coils, and therefore will affect the cooling coil loads (e.g., reduced return air enthalpy due to cool supply air leakage reduces mixed air enthalpy and therefore reduces cooling coil loads). To maintain the desired air pressure differentials across the building envelope, some return air is discharged outdoors. This means that some of the heating or cooling energy associated with leakage is discharged to outdoors and is not recaptured at the air-handler.

When the outdoor air temperature is between the desired supply air temperature and return air temperature high-limit set point, the economizer operates with 100% outdoor air and no return air enters the air-handler (all of the return air is discharged outdoors). In this case, even though mechanical cooling is used as a supplement to maintain the desired supply air temperature, the change in return air enthalpy due to duct leakage does not affect mixed air enthalpy or cooling coil loads. When the outdoor air temperature is below the desired supply air temperature, there is no mechanical cooling and duct leakage again has no impact on air-handler coil loads. However, to maintain the desired supply air temperature in this case, a change in return air temperature (e.g., due to duct leakage) will cause the economizer to alter the amounts of return air and outdoor air that enter the air-handler.

In the case of a VAV box with leaky downstream ducts, the duct leakage means that insufficient heating or cooling energy is delivered to the conditioned spaces. As a result, the thermostat call for heating or cooling is not satisfied and the thermostat calls for more air to be supplied through the VAV box. To deliver more supply air, the VAV box primary air damper opens further, which in turn reduces the resistance to airflow in the duct system. Consequently, to maintain the main duct static pressure at its set point, the supply fan airflow must increase to compensate for the downstream leakage airflows. Upstream leakage has a similar effect on supply fan airflow, but no effect on VAV box flows (unless the supply fan is too small to maintain duct static pressure in the leaky duct system).

Because the relationship between fan power and airflow is somewhere between a quadratic and cubic function (Wray 2003), the increase in supply airflow to compensate for duct leakage means that supply fan power consumption increases significantly, with a large fraction of this fan power used just to move the leaking air. Increasing the fan power also increases cooling coil loads when mechanical cooling is being used to maintain the desired supply air temperature (when the economizer is operating at 100% or minimum outdoor air). This occurs because the heat created by the increased fan power tends to increase the supply air temperature downstream of the fan. In response, the cooling coil water valve open furthers to provide more cooling to maintain the desired supply air temperature.

3. MODELING APPROACH

To evaluate the impacts of duct leakage on VAV system performance in large office buildings, we modeled a prototypical office building with different characteristics that represent three building vintages in three California climates with six different duct leakage configurations (54 cases), using DOE-2.1E (Winkelmann et al. 1993a, 1993b) and TRNSYS (Klein et al. 1996). Our modeling approach involves a three-step quasi-steady-state process in which the distribution system simulation is uncoupled from the loads and plant simulations of DOE-2, in the same manner that DOE-2 itself uses. The difference is that the TRNSYS system simulation expands beyond DOE-2 modeling capabilities to offer more flexibility in modeling duct thermal performance. The three steps in our modeling approach are as follows:

1. Hourly zone loads (heat extraction and addition rates) and zone air temperatures are calculated using DOE-2, for a constant air volume (CAV) system that has no duct leakage. These results are then output to a data file, which is read as input by TRNSYS. The data file also includes the corresponding hourly weather conditions, latent heat gains in conditioned spaces, and heat input to the ceiling plenum from lights. DOE-2 simulates all 8760 hours in a year.
2. TRNSYS generates hourly HVAC system fan and coil energy consumption data using interconnected detailed component models for the heating and cooling coils, fans, ducts, terminal boxes, economizer, and return plenum. The solution for each hour involves numerous iterations that terminate when convergence is achieved; convergence occurs when the error tolerances associated with component input and output variables are satisfied. Various duct leakage configurations are modeled at this stage. The TRNSYS analysis considers only hours when the HVAC system is operating. These hours (as defined in Appendix I) are: Monday through Friday, 6 a.m. to 8 p.m., and Saturday, 6 a.m. to 3 p.m.; they exclude Sundays and holidays (system is off on these days).
3. Regression analyses based on correlations that we developed between DOE-2 system and plant energy use are used to translate the TRNSYS system level coil load data to plant level energy use; energy costs are subsequently calculated based on this energy use.

In our evaluation, all but two of the effects described in Section 2.2 were modeled. The VAV box induction flows, as well as the impact on conditioned space loads of plenum temperature changes caused by duct leakage, were not modeled. Modeling these effects requires the use of coupled zone load and HVAC system models, which are not yet available in simulation tools that address duct leakage. Wray (2003) describes our modeling approach in more detail, the duct performance principles on which it is based, and the TRNSYS component models that we used.

An advantage of using the DOE-2/TRNSYS approach in this project is that DOE-2 prototypical models for a large commercial California building are already available, as are the custom TRNSYS component models (Franconi 1999). Another advantage is that the duct leakage modeling approach and its results for a California building have already been validated by Franconi, and no substantial changes to the simulation tool are required to carry out our analyses. No other whole-building modeling approach to assess duct system performance for large commercial buildings is currently as advanced as this approach.

4. BUILDING AND HVAC SYSTEM CHARACTERISTICS

In this study, we modeled a ten story, 150,000 ft^2 office building. Each story has a floor area of 15,000 ft^2 and is divided into five zones: four 15-ft wide perimeter zones and one core zone. Each set of five zones has a ceiling plenum above them that serves as the return air plenum. The mechanical plant is located in a below-grade basement.

4.1 Building Envelope

We modeled three construction eras (1980s, 1990s, and 2005) in three California climates where large commercial buildings are common (Oakland, Pasadena, and Sacramento). The building envelope thermal characteristics are listed in Table 1 for the 1980s and 1990s era buildings and in Table 2 for the 2005 era building. The general characteristics of the 1980s and 1990s era buildings were determined in a previous study (Huang and Franconi 1999). The 2005 era building is based on the requirements of the proposed 2005 California Title 24 Nonresidential Energy Standards (CEC 2003a).

In each case, the intermediate floors are 4 in. thick lightweight (80 lb/ft^3) concrete slabs, covered with a carpet and fibrous pad. The basement floor is a 6 in. thick heavyweight concrete slab on top of soil. The exterior walls are 1 in. thick stone (140 lb/ft^3), 2 in. x 4 in. steel studs (16 in. on center), insulation in the wall cavities, and 5/8 in. thick sheet rock. Windows are double-glazed. The bottom of each ceiling return plenum (conditioned space ceiling) is 3/4 in. thick, 2 ft. x 4 ft. acoustic ceiling tiles laid in a steel T-bar frame. The roof assembly above the top story's ceiling return plenum consists of built-up roofing, 4 in. thick lightweight concrete, and insulation. The R-values and U-values that are listed in Tables 1 and 2 are for entire assemblies, not including air films.

Table 1. Building Envelope Characteristics - 1980s and 1990s Construction Based on Huang and Franconi (1999)

	1980s Construction	1990s Construction
Roof		
Assembly R-value (h·°F·ft^2/Btu)	13.1	14.5
Walls		
Assembly R-value (h·°F·ft^2/Btu)	3.1	6.6
Windows		
Assembly U-value (Btu/(h·°F·ft^2))	0.72	0.60
Relative Solar Heat Gain (RSHG)*	0.69	0.62
Shading Coefficient**	0.77	0.71
Window/Zone-Wall Area Ratio	40%	50%

* RSHG is a function of the solar heat gain coefficient ($SHGC_{win}$), the window orientation, and the size and position of overhangs. Because the prototypes modeled do not have overhangs, RSHG=SHGC.
** Shading Coefficient = SHGC/0.87 = RSHG/0.87.

Table 2. Building Envelope Characteristics – 2005 Title 24
Based on the Draft 2005 Title 24 Standards (CEC 2003a)

	Oakland (CZ 3)	Pasadena (CZ 9)	Sacramento (CZ 12)
Roof			
Assembly R-value (h·°F·ft^2/Btu)	20.9	12.9	20.9
Walls			
Assembly R-value (h·°F·ft^2/Btu)	5.4	5.4	5.7
Windows			
Assembly U-value (Btu/(h·°F·ft^2))	0.77	0.77	0.47
Relative Solar Heat Gain (RSHG)*			
North	0.61	0.61	0.47
Non-North	0.41	0.34	0.31
Shading Coefficient (SC)**			
North	0.701	0.701	0.54
Non-North	0.471	0.391	0.356
Window/Zone-Wall Area Ratio	40%	40%	40%

* The CEC 2005 Title 24 Standards specify a maximum Relative Solar Heat Gain (RSHG) as listed above. RSHG is a function of the solar heat gain coefficient ($SHGC_{win}$), the window orientation, and the size and position of overhangs. We used the RSHGs specified in the 2005 Title 24 Draft Standards. Because the prototypes modeled do not have overhangs, RSHG=SHGC.
** Shading Coefficient = SHGC/0.87 = RSHG/0.87.

4.2 Building Operating Characteristics

Table 3 lists the operating characteristics that we used to model the building prototypes in DOE-2. Schedules describing when these characteristics apply are listed in Tables I-1 through I-7 of Appendix I. These schedules are based on the draft 2005 Title 24 schedules (CEC 2003a).

Table 3. Building Operating Characteristics

	1980s*	1990s*	2005 Title 24 Draft Standard**
Infiltration (ach)			
HVAC System Operating	0	0	0
HVAC System Off	0.30$^+$	0.30$^+$	0.075***
Minimum Outside Air (cfm/person)	15	15	15
Occupancy (ft^2/person)	100	100	100
Lighting Intensity (W/ft^2)	1.8	1.3	1.1
Equipment Load (W/ft^2)	0.75	0.75	1.34

* Huang and Franconi (1999), Table 10.
$^+$ Huang (2003).
** CEC (2003a).
*** Based on 0.038 cfm/ft^2 of exterior wall area, as proposed in the 2005 Title 24 Draft (CEC 2003a).

Infiltration is assumed to be zero when the HVAC system is operating. When the HVAC system is off, the infiltration rate is assumed to be the air change rate listed in Table 3, as appropriate for each case. The “off hours” infiltration rate for the 1980s and 1990s era buildings (0.3 ach, Huang

2003) is about midway in the range reported by Grot and Persily (1986) for eight 1980s era office buildings that they tested (0.1 to 0.6 ach). The hourly outdoor airflow rates modeled during system operating hours are based on the hourly occupancy schedules and the outdoor airflow rate requirements per person specified in the draft 2005 Title 24 standard (CEC 2003a).

Of the heat generated by the light fixtures, 45% is transferred to the occupied zones; the remainder goes to the ceiling return plenum (Huang 2003).

4.3 Air-Handling System Description

The single-duct VAV terminal-reheat air distribution system that we modeled in TRNSYS includes an airside economizer, a cooling coil, a variable-speed supply fan, five pressure-independent VAV-boxes (each with a discharge reheat coil), a ceiling return air plenum, and a variable-speed return fan. The system serves the five building zones on a single floor: four perimeter zones and one core zone. It is assumed that identical systems serve each of the ten floors in the building.

The system economizer uses the following control strategy:

- When the outdoor air temperature is above the return air temperature high-limit set point (70ºF in Sacramento, and 75ºF in Oakland and Pasadena, CEC 2003a), the amount of outdoor air entering the air-handler is the minimum required for ventilation. The remainder of the mixed airflow entering the air-handler (same flow rate as the supply airflow) is return air. Mechanical cooling is used to maintain the desired supply air temperature. To maintain a zero air pressure differential across the building envelope, the amount of return air discharged to outdoors is the same as the amount of outdoor air entering the air-handler.
- When the outdoor air temperature is between the desired supply air temperature and return air temperature high-limit set point, the economizer operates with 100% outdoor air and no return air enters the air-handler (all of the return air is discharged outdoors). Mechanical cooling is used as a supplement to maintain the desired supply air temperature.
- When the outdoor air temperature is below the desired supply air temperature, there is no mechanical cooling. In this case, the economizer mixes appropriate amounts of return air and outdoor air to maintain the desired supply air temperature.

In all cases, the minimum outdoor air ventilation rate is set to correspond to a minimum outdoor airflow of 2,250 cfm per floor at design conditions. This value is based on the occupant density of 100 ft^2/person and the outdoor-air ventilation rate of 15 cfm/person described in Table 3. For each case, the minimum outdoor air ventilation rate is a constant fraction of the supply fan airflow, but this fraction is not necessarily constant from case to case because design supply airflows vary from case to case.

The cooling coil control is simple: a constant supply air dry-bulb temperature of 53ºF is maintained downstream of the supply fan. This temperature was selected to achieve a 20ºF supply air temperature difference relative to the 73ºF cooling set-point temperature of the conditioned spaces.

All VAV boxes have the same flow fraction at their minimum turndown. For each box, this fraction is set at 40% of the design maximum flow rate entering the box to ensure that sufficient heat can be delivered to the zone, assuming a 180°F water temperature entering the reheat coils. In some cases, lower turndown fractions (e.g., 30%) could have been used to satisfy heating requirements; however, for consistency, we used the same turndown fraction in all cases.

4.4 Cooling and Heating Plant Description

A water-cooled hermetic centrifugal chiller supplies chilled water to the air-handling system cooling coil. The chiller rejects heat outdoors using a cooling tower. A natural-gas-fired boiler supplies hot water to the VAV box reheat coils. We used the default DOE-2 plant equipment models for the chiller, cooling tower, boiler, and associated circulation pumps.

The heat gain associated with the boiler standby loss to the unconditioned basement (Btu/h) is calculated as 0.0057 times the boiler fuel efficiency (80% for the 2005 vintage, 79% for the others) times the total building occupied floor area (Franconi 1999). Combustion air to the basement for the boiler is assumed to be two air changes per hour (Huang 2003).

4.5 Duct Leakage Characteristics

Upstream Leakage

The supply and return fans in the VAV system have variable-speed-drive control. Although not modeled explicitly, we assume that the HVAC control system varies the supply fan airflow to maintain a constant duct static pressure upstream of the VAV boxes. In a VAV distribution system with constant static-pressure control, the pressure distribution along the ducts upstream of the VAV zone boxes is affected by several parameters, which include: the duct friction and fitting pressure drops, the system equipment (e.g., mixing dampers, cooling coil, air filters) pressure drops, the static-pressure set point, and the placement of the static-pressure sensor. The duct and system equipment pressure drops vary with airflow. Therefore, in general, the pressure differences across the upstream leaks when the fan operates at design conditions (maximum fan airflow) will differ from the pressure differences across the leaks during part-load fan operation (reduced fan airflow). In certain circumstances, upstream leakage airflow is not affected by part-load fan operation and the average upstream duct air leakage is constant. This is only precisely true when all of the duct leaks are located at the same location as the pressure sensor, and pressure reset control is not in use.

The simplifying assumption that we used for modeling leakage upstream of the VAV boxes is that the upstream leakage airflow is constant and is not affected by the airflow through the fan. This implies that the fraction of the fan airflow that is leaking upstream of the VAV boxes increases as the fan airflow is reduced.

Downstream Leakage

Downstream of a VAV box, the duct pressure distribution is affected by the box damper position, which provides a variable flow resistance to control the downstream duct airflow. The pressure differences across the leaks in the downstream ducts can be related to the average pressure drop through these ducts. If turbulent flow through the duct is assumed, the airflow rate affects the duct pressure drop according to the square law. If it is also assumed that there is a square root relationship between leakage flow and pressure difference across the duct leaks, then the fraction of the VAV box airflow that leaks from the ducts downstream of the boxes remains approximately constant. However, the leakage airflow is not constant.

Nominal Leakage Fraction

Based on the simplifying assumptions described above, two inputs are required to describe supply duct leakage in the TRNSYS simulation: upstream leakage fraction and downstream leakage fraction. The upstream leakage fraction is the upstream leakage flow, which is a constant for all part load ratios, divided by the supply fan design airflow. The downstream leakage fraction is a constant fraction of the VAV box airflow, which varies during system operation.

In the TRNSYS simulations, we used six leakage configurations in each of the three climates for each of the three building vintages (54 cases) to evaluate the variability of duct leakage impacts on HVAC system energy performance:

- 10+10, which refers to a 10% leakage fraction upstream of the VAV boxes and a 10% leakage fraction downstream of the VAV boxes (about 19% total leakage) at design flow;
- 7.5+7.5, which refers to 7.5% leakage fractions upstream and downstream (about 14% total leakage) at design flow;
- 10+2.5, which refers to a 10% leakage fraction upstream and a 2.5% leakage fraction downstream (about 12% total leakage) at design flow;
- 2.5+10, which refers to a 2.5% leakage fraction upstream and a 10% leakage fraction downstream (also about 12% total leakage) at design flow;
- 5+5, which refers to 5% leakage fractions upstream and downstream (about 10% total leakage) at design flow; and
- 2.5+2.5, which refers to 2.5% leakage fractions upstream and downstream (about 5% total leakage) at design flow.

The last case represents a tight duct system, but not a perfect one with zero leakage. It is unlikely that real duct systems can be made perfectly tight.

Note that the sum of the upstream and downstream leakage fractions at design flow do not equal the total leakage fraction. This is because the upstream leakage is a fraction of the supply fan flow and the downstream leakage is a fraction of the flow entering the VAV boxes. For example, in the 10+10 case, if the supply fan flow is 10,000 cfm, then the upstream leakage is 1,000 cfm (10% of 10,000 cfm) and 9,000 cfm reaches the VAV boxes. The downstream leakage is therefore 900 cfm (10% of 900 cfm) and 8,100 cfm reaches the zones. This means that a total of 1,900 cfm or 19% of the 10,000 cfm supply fan flow has leaked from the ducts.

4.6 Plenum Energy Balance

In our TRNSYS model of the ceiling return air plenum, the zone return air passes through an open ceiling plenum and then to the return air ducts and fan. An energy balance is used to determine the return plenum air temperature. This energy balance accounts for the effects of supply-duct air leakage, plenum “floor” (zone ceiling) and “ceiling” (zone floor) conduction, plenum exterior wall conduction, heat gain from ceiling-mounted lights, and zone return airflow.

Our simulations show that the plenum is slightly cooler when there is duct leakage. For each hour in the leakiest case (19% total duct leakage), the plenum temperature is 1 to 2ºF cooler than the corresponding temperature in the “tight” (5% total duct leakage) case. The largest plenum temperature reduction occurs when the cooling effect due to supply air leakage is largest, which is also when the largest net cooling load in the conditioned zones occurs. These plenum temperature changes are consistent with our field observations in an office building when 15% leakage was added to a VAV system with 5% leakage (Diamond et al. 2003).

Although we included the effects of plenum "floor" and "ceiling" conduction in calculating the return plenum air temperature, our uncoupled sequential approach to evaluate the zone loads and HVAC system performance ignores the impact of the plenum temperature changes due to leakage on heating or cooling loads and air temperatures in the conditioned zones. We ignored this effect because it is small compared to the impacts of other gains and losses in the conditioned spaces (e.g., solar loads; occupancy, equipment, and lighting heat gains; exterior wall and window conduction). For example, the largest plenum temperature reduction (2ºF) due to 19% total leakage, which corresponds with the largest net cooling load in the conditioned zones, would only reduce the cooling load by about 3%. A more rigorous approach to account for this effect would involve a coupled simultaneous solution of the loads, system, and plant performance. In the future, EnergyPlus could be used for this purpose if the TRNSYS duct models were integrated with that program.

4.7 Fan Performance

In many hourly simulation programs, including DOE-2, the fan performance subroutines are based on a third-order polynomial relating the fan fractional shaft power to the fan part load airflow ratio (Brandemuehl et al. 1993). The form of the equation is:

$$FPR = c_0 + c_1 \cdot PLR + c_2 \cdot PLR^2 + c_3 \cdot PLR^3 \qquad (1)$$

where

- *FPR*: Fan power ratio, which is the dimensionless ratio of the fan shaft power at a particular time to the fan shaft power under design conditions;
- *PLR*: Part load ratio, which is the dimensionless ratio of the fan airflow at the same time to the fan airflow under design conditions; and
- $c_0 \ldots c_3$: Constant coefficients for the curve fit. The specific coefficients depend on the pressure drop, pressure control, and airflow characteristics of the system.

Table 4 defines the coefficients for various fan control schemes. These include: outlet damper control, inlet vane control, and variable speed control. There are two sets of coefficients listed for variable speed control. One is a set of coefficients used in DOE-2 and in the ASHRAE HVAC Toolkit for a generic fan, and produces a curve similar to the one used by Franconi (1999) for part load airflow fractions of one or less. The other set corresponds to the relation defined in the Title 24 Nonresidential Alternative Calculation Method (ACM) (CEC 2003b) for a variable speed drive with static pressure control. We used the Title 24 set of coefficients in our simulations.

Table 4. Polynomial Coefficients for Fan Performance Curves

Fan Control Type	c0	c1	c2	c3
Outlet Damper	0.3507	0.3085	-0.5414	0.8720
Inlet Vane	0.3707	0.9725	-0.3424	0
Variable Speed Drive (Generic)	0.0015	0.00521	1.1086	-0.1164
Variable Speed Drive (Title 24)	0.1021	-0.1177	0.2647	0.7600

Figure 1 shows the differences between the relationships. For fan part load airflow fractions greater than about 0.33, the Title 24 curve results in the lowest fan power. In our simulations, fan part load flow fractions were typically concentrated in a range of 0.4 to 0.8.

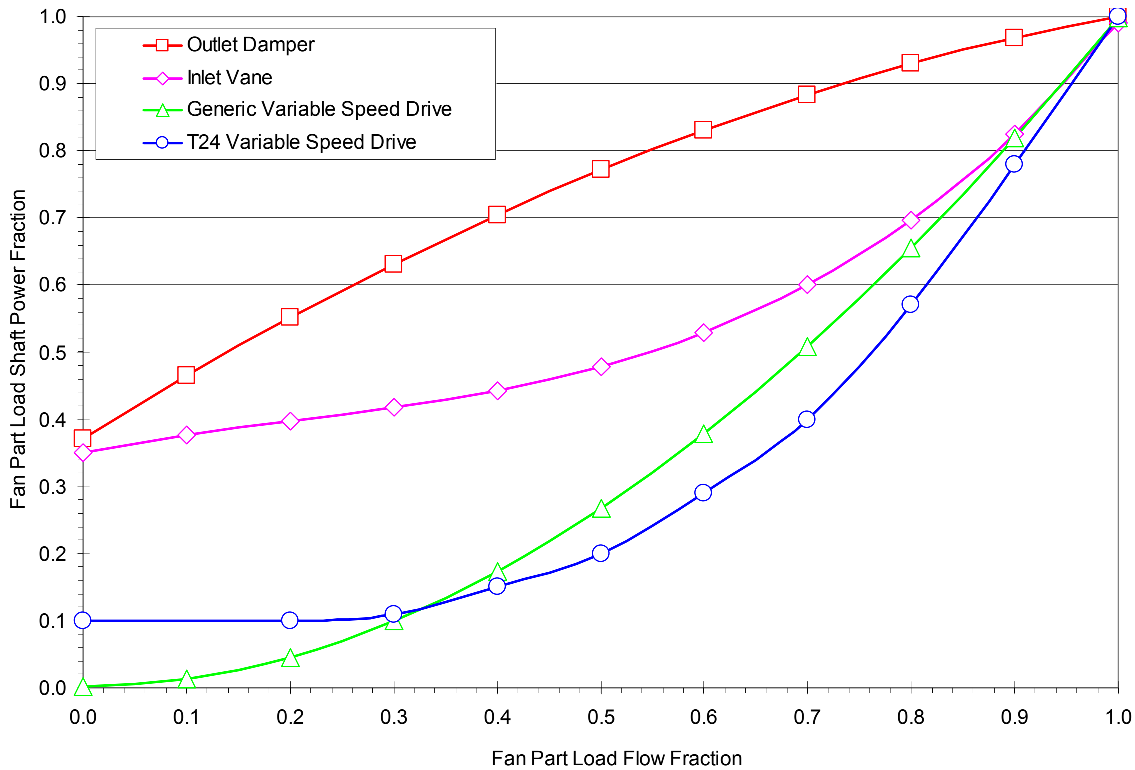

Figure 1. Comparison of Fan Performance Curves

4.8 System and Plant Sizing

The VAV system that we simulated in TRNSYS used the same size system and plant equipment for the various duct leakage cases in a given climate and for a given building vintage; however, the sizes varied over the nine building vintage and climate combinations.

The supply (and return) fan design airflow was determined by the high-leakage case (10+10), because the maximum airflow occurs for that case. The intermediate-floor supply fan design airflows for each climate and building vintage combination are listed in Table 5, and are based on the calculated zone airflow requirements with leakage effects added. Supply and return fan power at design conditions are based on the design airflow, total pressure rises of 3 in. of water for the supply fan and 1 in. of water for the return fan, a combined fan and drive efficiency of 65% for each fan, and motor efficiencies of 90% for the supply fan and 88% for the return fan. Based on these fan parameters, the specific total fan electrical power is 0.8 W/cfm. These parameters represent a low-pressure system that serves a single floor. Systems with larger pressure rises will use more fan power, which will make duct sealing even more cost-effective.

Table 5. TRNSYS Air-Handler Fan Design Parameters (Airflows and Electrical Power for Intermediate Floor)

Climate Zone	Vintage	Supply Fan Airflow (cfm)	Supply Fan Power (kW)	Return Fan Power (kW)	Total Fan Power (kW)
CZ3	1980s	13,000	7.8	2.8	10.6
(Oakland)	1990s	13,000	7.8	2.8	10.6
	2005	10,500	6.3	2.2	8.5
CZ9	1980s	21,100	12.7	4.5	17.2
(Pasadena)	1990s	23,700*	14.3	5.0	19.3
	2005	15,200	9.1	3.2	12.3
CZ12	1980s	16,800	10.1	3.6	13.7
(Sacramento)	1990s	16,300	9.8	3.5	13.3
	2005	12,100	7.3	2.6	9.9

* Increased wall insulation and window solar heat gain in the 1990s changed the time (and therefore outdoor conditions) when peak loads occur in the Pasadena building. This resulted in increased indoor temperatures when the air-handling system is off, which in turn resulted in larger cooling loads at the start of the occupied (conditioned) periods.

The chilled-water coil and VAV-box reheat coils are also sized sufficiently to meet the maximum coil loads (20% oversizing). For sizing the cooling coils, we assumed a 12ºF water-side temperature rise and an entering water temperature of 44ºF; for the reheat coils, we assumed the water-side temperature drop was 30ºF and the entering water temperature was 180ºF.

Table 6 summarizes the cooling and heating coil sizes per floor that were generated by DOE-2 (for a CAV system), and which DOE-2 used to size the plant equipment for its plant energy use simulations (with no duct leakage). Table 6 also lists the corresponding coil sizes that we calculated and that were used in the TRNSYS VAV system simulations. The TRNSYS coil sizes differ from the DOE-2 sizes for three reasons:

1. The TRNSYS cooling and heating coil sizes account for the effects of duct leakage on coil loads.
2. The TRNSYS reheat coil sizes are for a VAV system rather than a CAV system, and VAV system reheat loads are smaller because supply airflows are lower during reheat for a VAV system.
3. The TRNSYS sizes are based on the zone loads and corresponding zone temperatures generated by DOE-2, but are determined using VAV system-sizing calculations independent of DOE-2. The calculations that we used are based on methods outlined by Knebel (1983), Kreider and Rabl (1994), and Pedersen et al. (1998).

Table 6. Cooling and Heating Coil Sizes (kBtu/(h·floor))

		DOE-2		TRNSYS	
Climate Zone	Vintage	Cooling	Heating	Cooling	Heating
CZ3 (Oakland)	1980s	315	354	470	262
	1990s	308	331	458	236
	2005	252	271	407	202
CZ9 (Pasadena)	1980s	428	277	528	177
	1990s	419	270	544	185
	2005	353	198	483	129
CZ12 (Sacramento)	1980s	450	438	598	337
	1990s	434	397	576	294
	2005	339	249	483	180

5. SYSTEM - PLANT ENERGY USE REGRESSIONS

As described in Appendix II, we determined that the whole-building heating and cooling plant hourly demands and annual energy consumption (electricity and natural gas) can be predicted based on the heating and cooling coil loads of a mid-height intermediate floor. In this analysis, we calculated hourly heating and cooling part load factors for the intermediate floor and for the whole building. For the intermediate floor, the five reheat coil loads were summed for each hour to obtain an hourly total heating coil load for that floor. The hourly total heating coil loads for the floor were then divided by the maximum of those values to obtain the intermediate-floor hourly heating part load ratios. We used the hourly total cooling coil loads for the same floor in a similar manner to determine the intermediate-floor hourly cooling part load ratios. Also, we used the whole-building hourly heating and cooling total coil loads in the same manner to obtain the whole-building heating and cooling part load ratios.

We used regression techniques to generate polynomial relationships between the intermediate-floor hourly part load ratios and the hourly whole-building plant energy demand (chiller electricity, cooling tower electricity, and boiler electricity and natural gas). Tables II-3a though II-6c in Appendix II provide the regression equations, equation coefficients, regression R^2 values, and example predicted values. The R^2s for the regression equations for all three vintages and climate zones ranged from 0.9999 to 1.000 for the chiller electricity demand, 0.9338 to 0.9997 for the cooling tower electricity demand, 0.9990 to 1.0000 for the boiler electricity demand, and 0.9984 to 0.9997 for the boiler natural gas demand. The resulting equations were applied to the TRNSYS coil loads to predict whole building plant electricity and natural gas consumption for each of the various leakage cases modeled.

Figures 2 through 7 provide example regression plots to illustrate the relationships between the various parameters for the 2005 Title 24 compliant building in Sacramento (CEC Climate Zone 12). These plots are representative of the plots for other climate zones and building vintages. In particular, Figures 2 and 3 compare the whole-building part load ratios and the intermediate-floor part load ratios. Figures 4 through 7 show, for the same building prototype and climate, the chiller, cooling tower, and boiler electricity demand curves, and the boiler natural gas demand curve, all based on the intermediate-floor part load ratios. Compared to the other plant demand data, the cooling tower electricity data has more scatter. However, the annual cooling tower electricity consumption predicted using the regression equation was less than 1% different from the annual sum of the cooling tower electricity consumption reported by DOE-2.

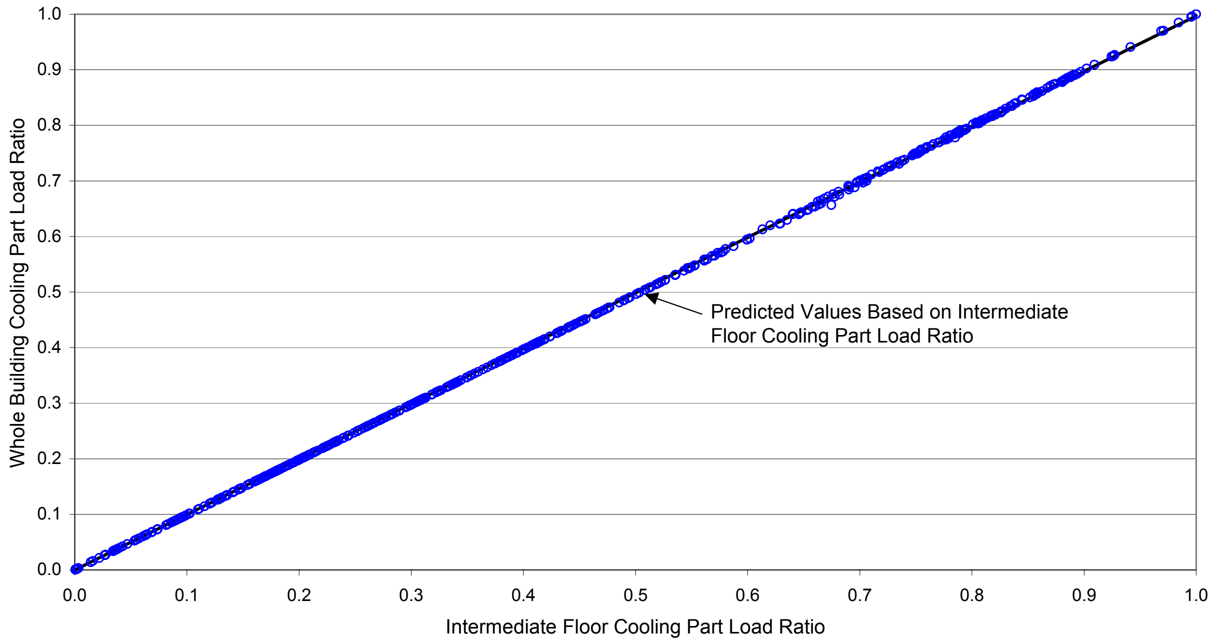

Figure 2. Building Cooling Part Load Ratio Regression
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

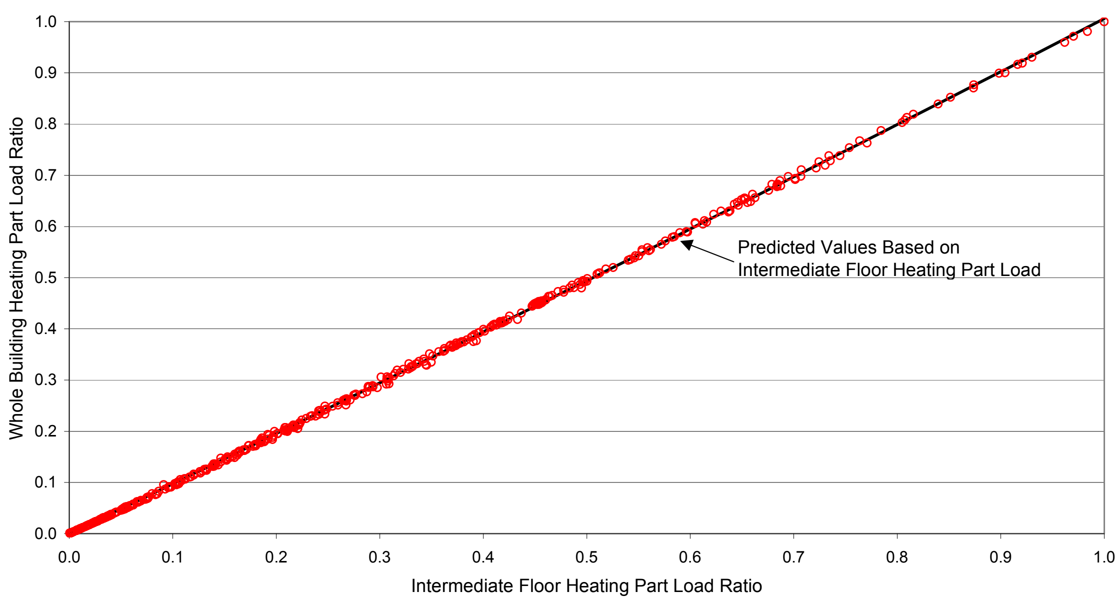

Figure 3. Building Heating Part Load Ratio Regression
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

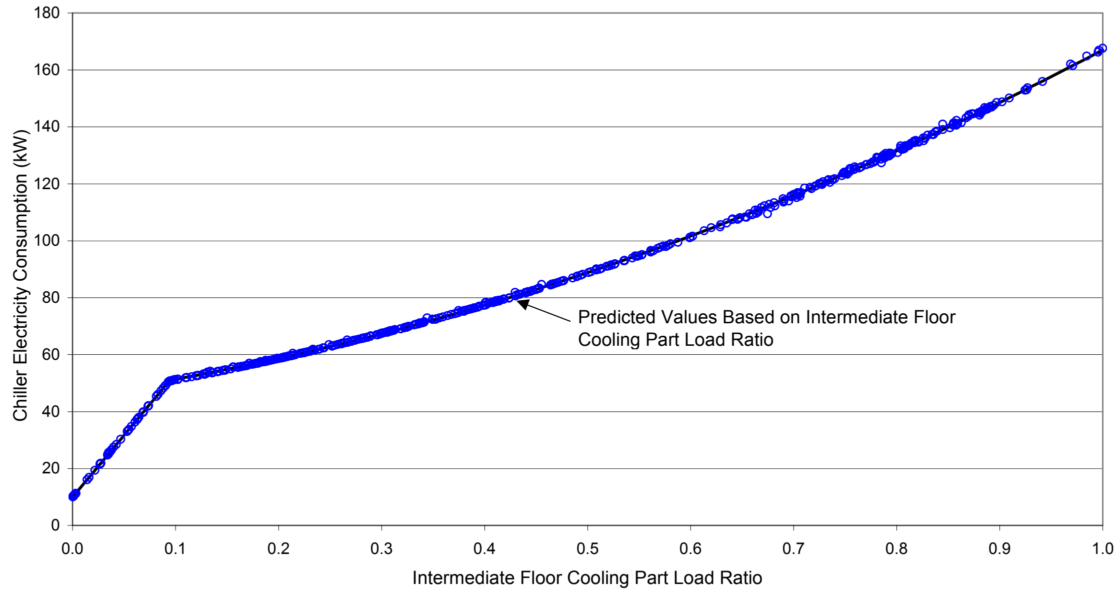

Figure 4. Chiller Electricity Consumption (kW)
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

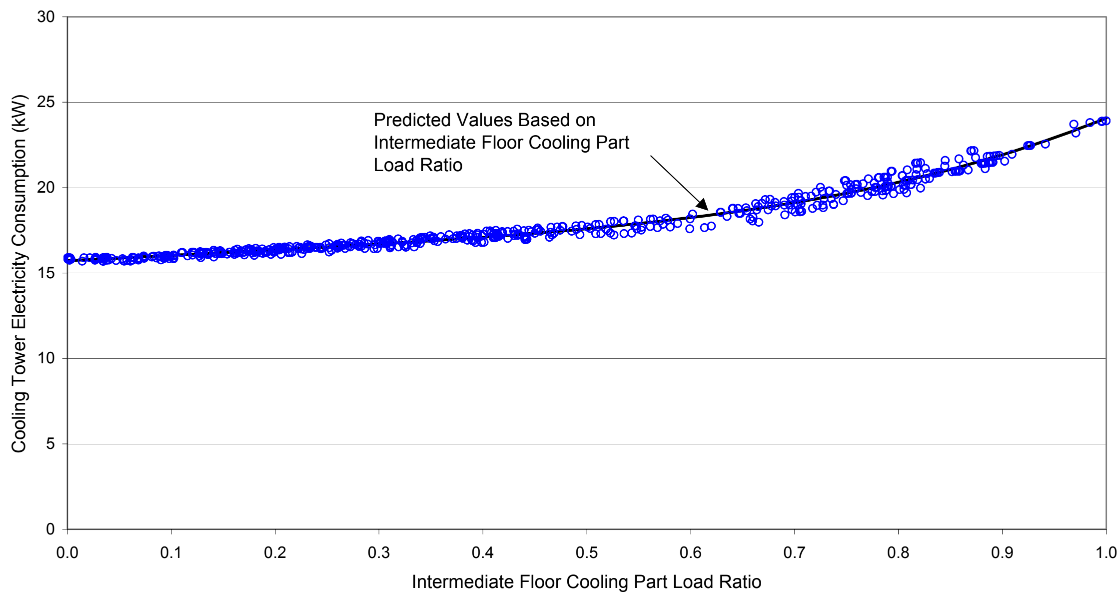

Figure 5. Cooling Tower Electricity Consumption (kW)
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

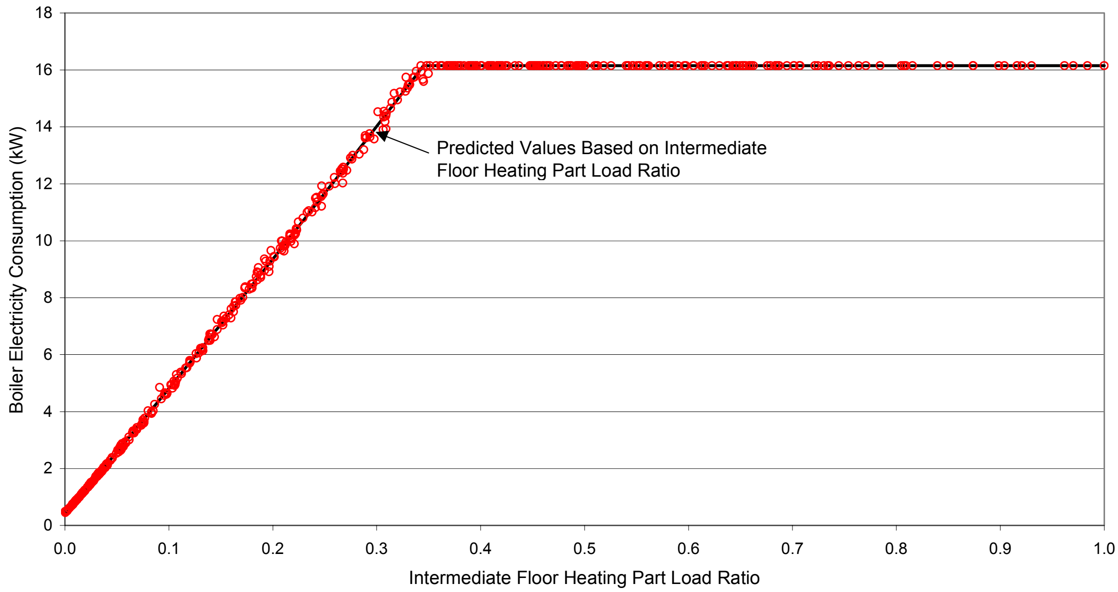

Figure 6. Boiler Electricity Consumption Regression (kW)
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

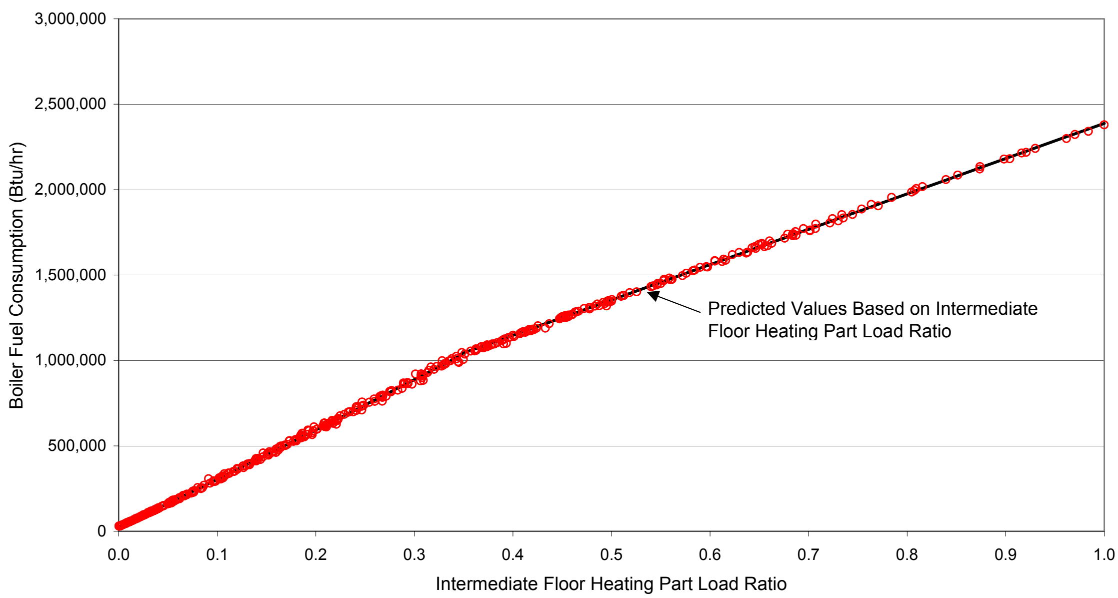

Figure 7. Boiler Fuel Consumption Regression (Btu/h)
CZ 12 (Sacramento) - 2005 Title 24 Compliant Large Office Building

6. RESULTS

The sequential DOE-2/TRNSYS modeling approach could best be described as "user hostile". It is unlikely this approach would be practical on a day-to-day basis for Title 24 compliance analyses or even ever used outside a research environment. To reduce the difficulty of using this approach, the authors have developed a spreadsheet-based "graphical interface" (not generated by TRNSYS itself) that organizes and displays various key input and output parameters of TRNSYS. This graphical aid greatly facilitates understanding the complex interactions between system flows, loads, and temperatures.

Figures 8 and 9 show two samples of the performance parameters calculated by TRNSYS for two different hours of VAV system operation in the 2005 Title 24 Sacramento building. Both cases represent a system with 10% duct leakage upstream and 10% duct leakage downstream of the VAV boxes at design conditions. Dashed lines leading from the ducts to the ceiling plenum show leakage paths.

Figure 8 shows the system performing on a cool January day under heating conditions, with the VAV boxes operating at or near their minimum flows, and with reheat being added to the supply air for all five zones. In this case, the economizer is partly open to mix outdoor air with return air and maintain the desired supply air temperature downstream of the supply fan, so that no heat needs to be extracted by mechanical cooling through the cooling coil. A supply air temperature reset strategy would reduce the reheat coil loads in this case, but our TRNSYS models for a VAV system do not include this capability.

In Figure 9, the system is performing with a large cooling load in every zone at the start of a warm July day. All VAV boxes are open part way to supply sufficient cool air to meet the zone loads. There is no reheat in this case. The economizer is open completely to reduce the mechanical cooling through the cooling coil. All return air is exhausted to outdoors.

6.1 Air-Handler Fan Power Ratios

The largest effect that duct leakage has on distribution system performance is to increase fan energy consumption. Using the DOE-2/TRNSYS simulation approach, we explored the impacts of upstream and downstream leakage independently and in combination.

Figures 10 through 14 show the hourly supply and return fan power ratios versus the fraction of design airflow delivered to the zones for the 2005 Title 24 Sacramento building. The fan power ratio is the hourly fan power for the leaky duct case relative to the fan power in the same hour for the tight duct system (about 5% total leakage). Five cases are shown: 2.5% upstream leakage plus 10% downstream leakage, 10% upstream leakage plus 2.5% downstream leakage, 10% upstream leakage plus 10% downstream leakage, 7.5% upstream leakage plus 7.5% downstream leakage, and 5% upstream leakage plus 5% downstream leakage. The upstream leakage is a fixed mass flow (specified fraction of supply fan design flow rate); the downstream leakage is a fixed fraction of VAV box flow, even under part-load conditions. The air mass flow through the return fan matches the air mass flow through the supply fan (return fan and supply fan volumetric flows differ due to air temperature differences between the two airstreams).

Plots for other climates and building vintage combinations are not shown, but are similar to the five included here.

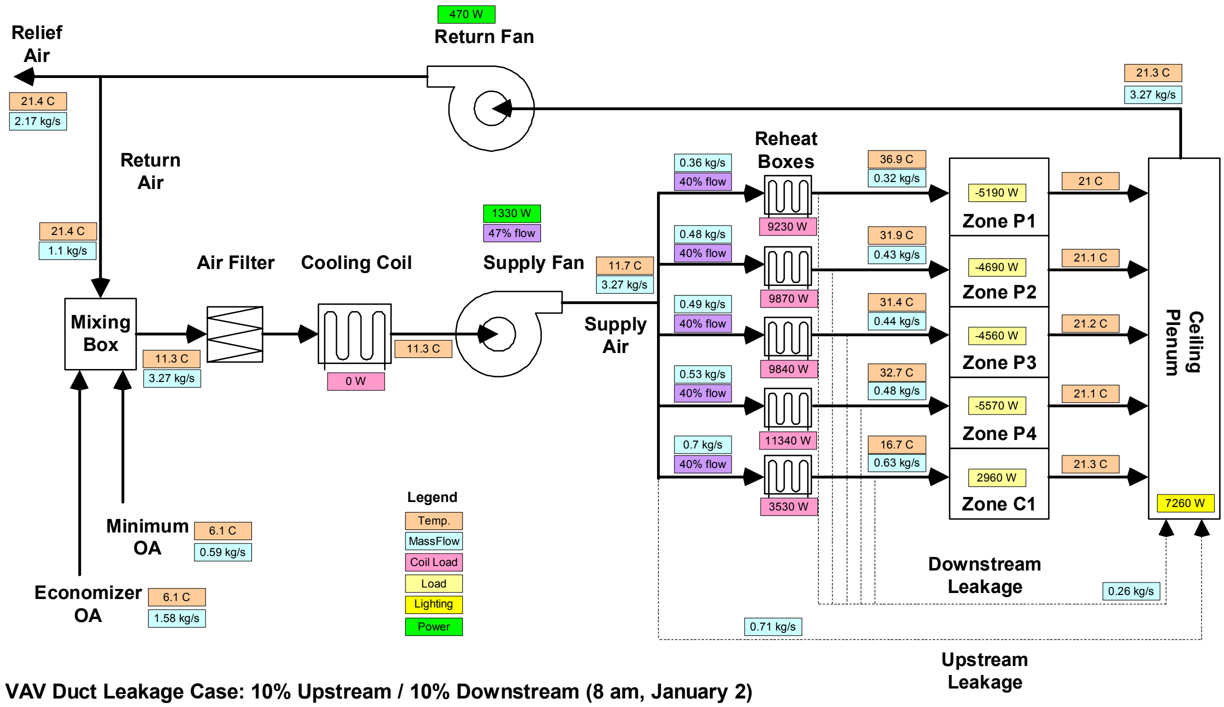

Figure 8. Sample TRNSYS Output – Heating Hour

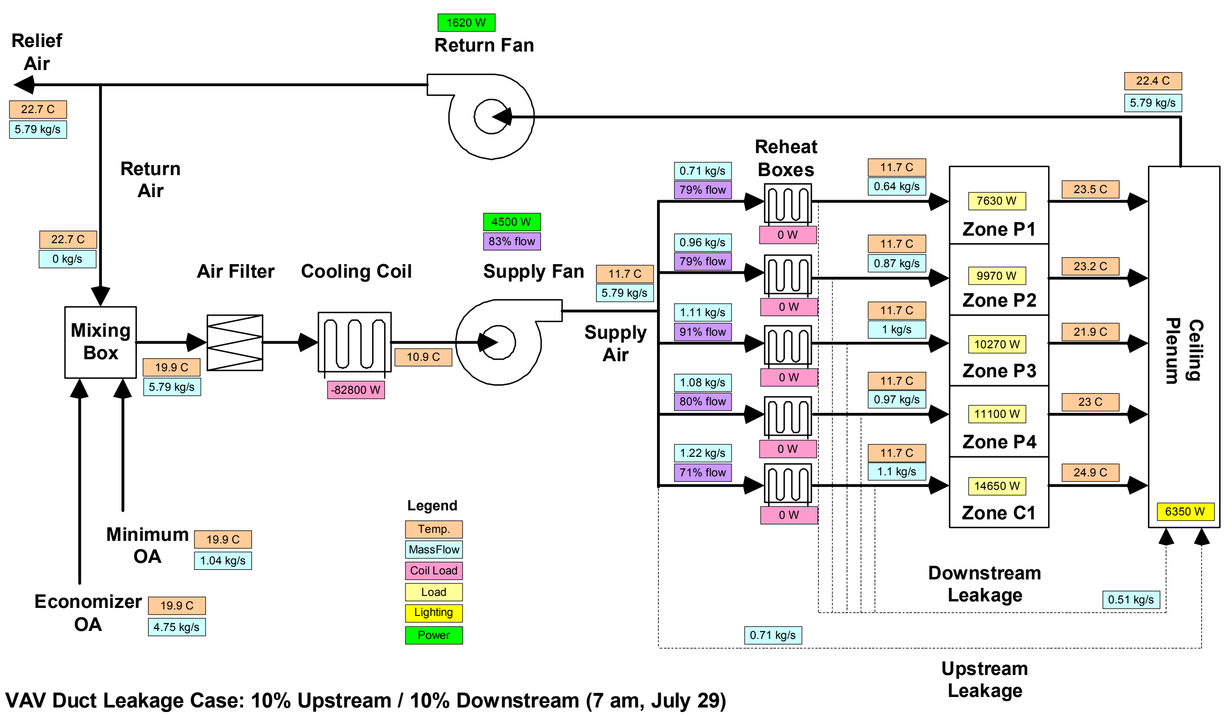

Figure 9. Sample TRNSYS Output – Cooling Hour

Dominant Downstream Leakage

The effect on fan power of only increasing the downstream leakage is shown in Figure 10. In this case, the downstream leakage is increased from 2.5% leakage to 10% leakage, while the 2.5% upstream leakage remains unchanged. The total leakage with the increased downstream leakage is about 12%.

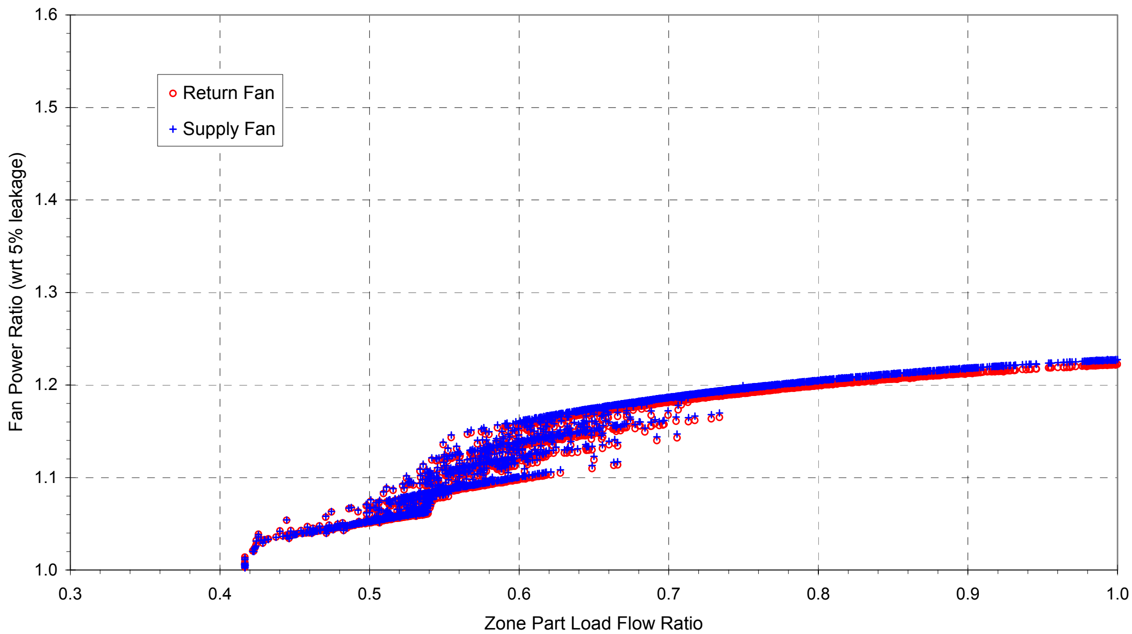

Figure 10. Dominant Downstream Leakage (2.5+10) - Fan Power Impacts

Compared to the tight duct system (5% total leakage) at design conditions (zone part load flow of 1.0), Figure 10 shows that the added downstream leakage increases supply and return fan power about 23%. The fan power increases are reduced as the zone-part-load-flow ratio decreases and, at part loads less than about 0.73, the curves become quite scattered. At the average zone-part-load-flow ratio (0.65), the power increases for both fans are in a broad range of about 11 to 18%. Because the two fans behave similarly, the fractional increase in total fan power is similar to the average fractional increase for the supply and return fans at any particular zone-part-load-flow ratio. The average increase in total fan power for this dominant downstream leakage case is about 14%.

The supply fan power ratios increase as the zone part load flow ratios increase, because as Figure 11 shows, the downstream leakage airflow and therefore supply fan airflows increase more with increasing part load than for the tight duct system (the downstream leakage is a fixed fractional flow, but not a fixed flow rate). Because the return fan mass flow (not shown in Figure 11) is the same as the supply fan mass flow, the return fan power ratios increase in a similar manner.

The scatter at a given zone-part-load-flow ratio occurs because there are some hours when no supply air reheating is needed and all the VAV boxes are supplying more than their minimum turndown flow, and there are other hours at the same zone-part-load-flow ratio when one or more of the zones requires reheat and the corresponding VAV boxes are providing only the minimum turn down flow. In the latter circumstance, because the VAV box airflow is constant, increased leakage flows downstream of these boxes do not increase the supply and return fan airflows, and therefore the leakage downstream of these boxes does not increase fan power. However, for the other VAV boxes that are not at their minimum turndown, increased leakage flows downstream of these boxes do increase the supply and return fan airflows, and therefore do increase fan power.

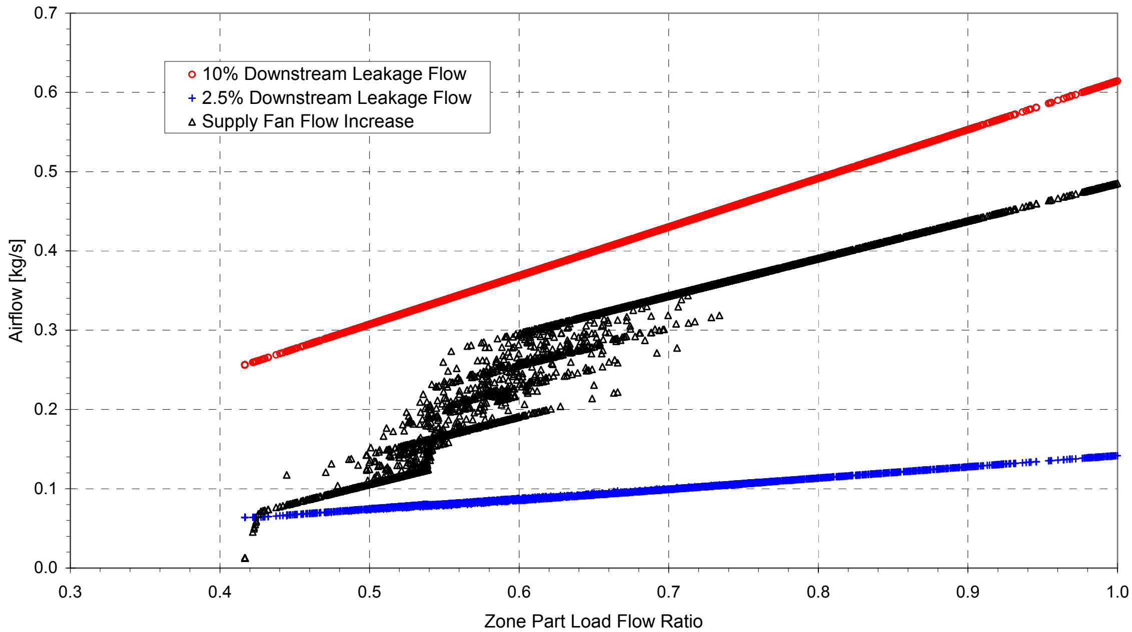

Figure 11. Dominant Downstream Leakage (2.5+10) - Airflow Impacts

To illustrate the behavior when there is some reheating, Figures 12 and 13 show a sample of the performance parameters calculated by TRNSYS for a cooling hour with reheat during January at the average zone-part-load-flow ratio (0.65), for the 2005 Title 24 Sacramento building. Figure 12 shows a tight duct system (about 5% total leakage); Figure 13 shows the same system, but with leaky downstream ducts (about 12% total leakage). In this example, the increase in total fan power with the increased downstream leakage is about 11%.

A positive consequence of downstream duct leakage is that the amount of reheating required will be reduced for the leaky system, because the supply airflows entering the zones with reheat are less than for the tight system and less overcooling will occur due to the airflow entering the zone. This consequence of downstream leakage actually causes system reheat loads to decrease slightly, as shown in Figures 12 and 13 and as noted in the annual energy consumption comparisons discussed in Section 6.2. On the other hand, it is worth noting that some zones do not receive their required minimum outdoor air through the HVAC system for the leaky duct case.

Dominant Upstream Leakage

The effect on fan power of only increasing the upstream leakage is shown in Figure 14. In this case, the upstream leakage is increased from 2.5% leakage to 10% leakage, while the 2.5% downstream leakage remains unchanged. The total leakage with the increased upstream leakage is about 12%.

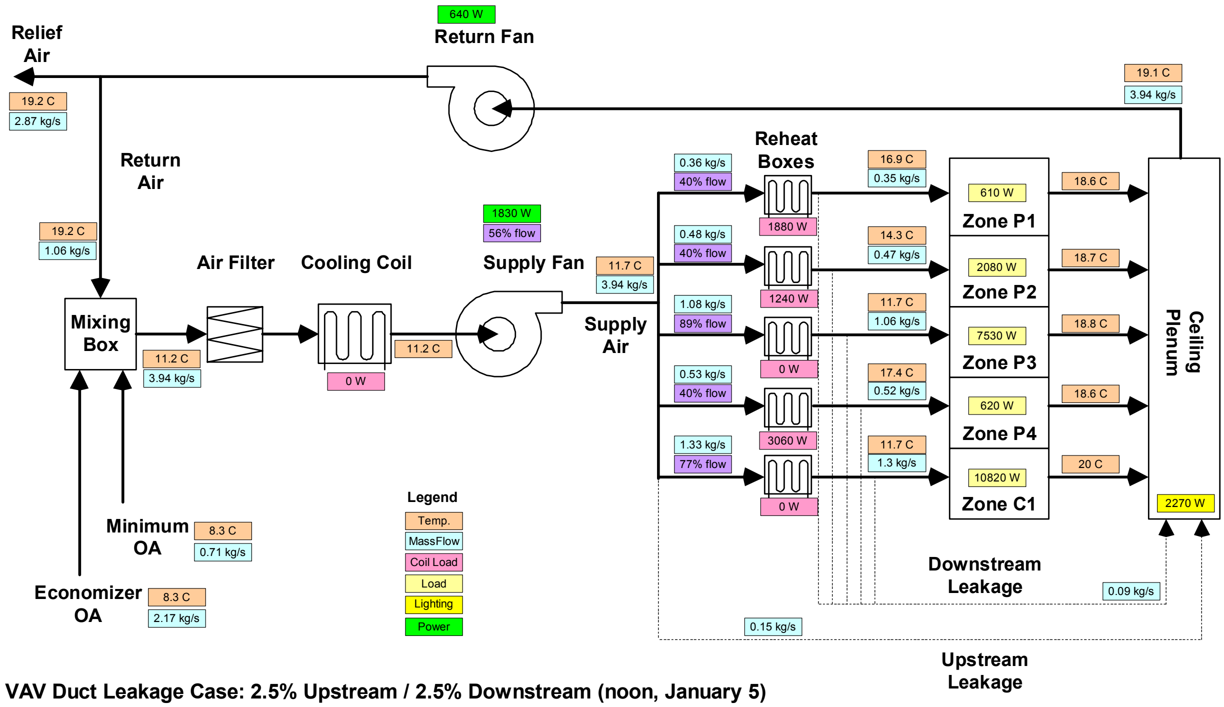

Figure 12. Tight Ducts (2.5+2.5) – Cooling Hour with Reheat

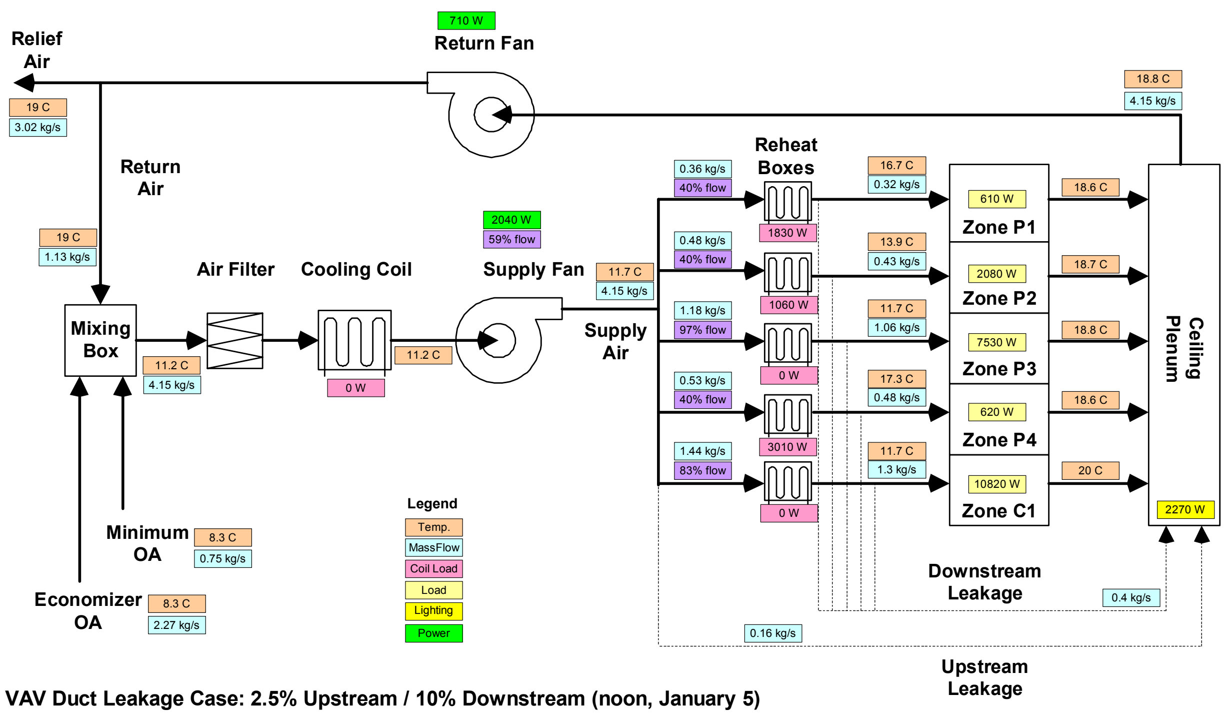

Figure 13. Dominant Downstream Leakage (2.5+10) – Cooling Hour with Reheat

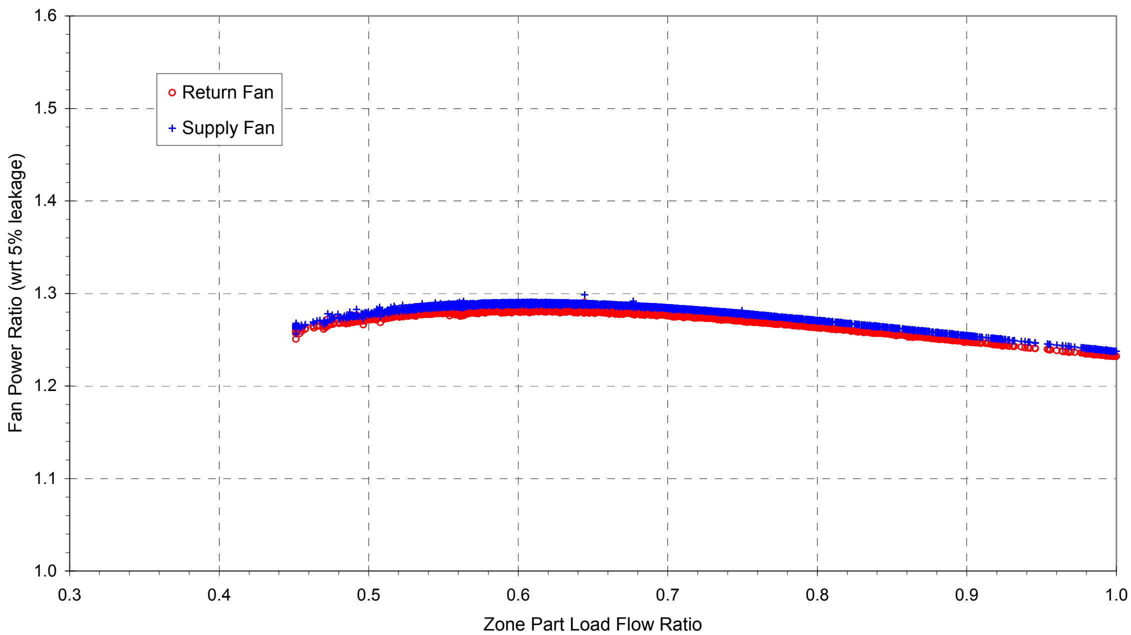

Figure 14. Dominant Upstream Leakage (10+2.5) - Fan Power Impacts

Compared to the tight duct system at design conditions, Figure 14 shows that the added upstream leakage increases supply and return fan power about 24%. At the average zone-part-load-flow ratio (0.66), the power increases about 29% and 28% respectively for the supply and return fans. The average increase in total fan power for this dominant upstream leakage case is about 28%.

The behavior with dominant upstream leakage is very different compared to the behavior for dominant downstream leakage. With a fixed leakage rate, the upstream leakage flow becomes a larger percentage of total airflow at lower zone-part-load-flow ratios. As a result, in the absence of downstream leakage, the fan power ratio would continually increase as the part load was reduced. However, the 2.5% downstream leakage in this case reduces the fan power ratio as the part load reduces, and the net effect is as shown in Figure 14.

Combined Upstream and Downstream Leakage

Figure 15 shows the results for the 10+10 leakage case (about 19% total leakage), which combines the separate effects of dominant upstream leakage and dominant downstream leakage on fan power consumption. Overall, the increase in fan power due to the combined leakage is greater at all zone-part-load-flow ratios in this case than in either the dominant downstream or dominant upstream leakage cases described earlier. Compared to the tight duct system at design conditions, the supply and return fan power increase about 53% and 51% respectively. At the average zone-part-load-flow ratio (0.65), the total fan power increase due to leakage ranges from 45 to 54%. The average increase in total fan power for this combined leakage case is about 50%.

Figures 16 and 17 show the results for the 7.5+7.5 and 5+5 leakage cases (about 14% and 10% total leakage respectively). Compared to the 10+10 case, as the downstream leakage fractions decrease, the scatter decreases significantly; the impact on reheat coil loads also decreases significantly. The average increase in total fan power for these two cases is about 30% and 13% respectively.

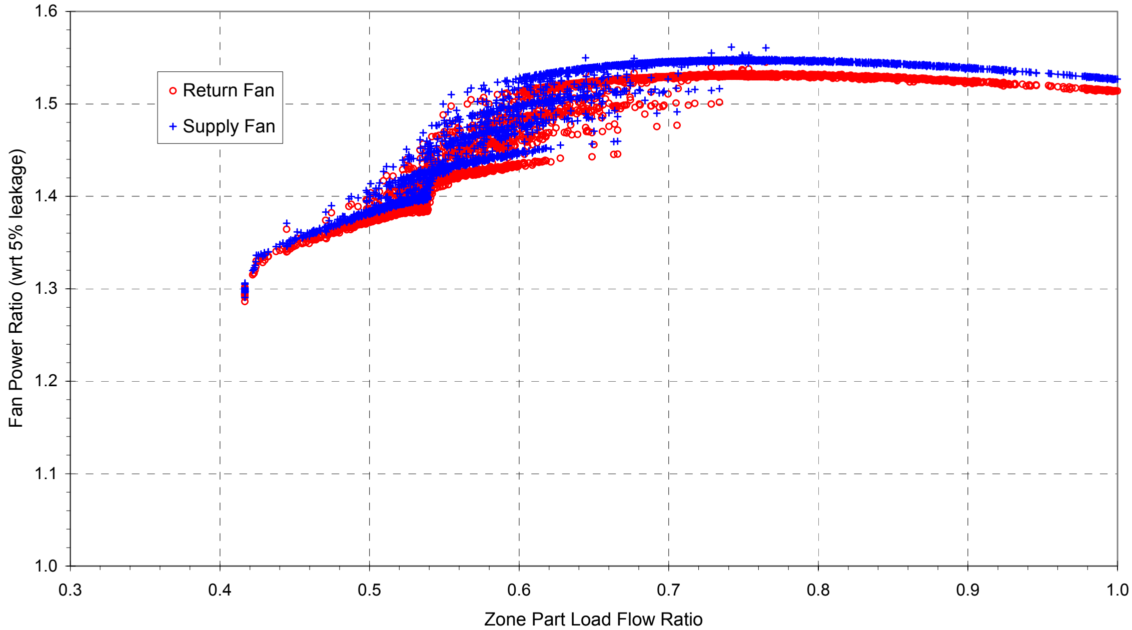

Figure 15. Upstream and Downstream Leaks (10+10) - Fan Power Impacts

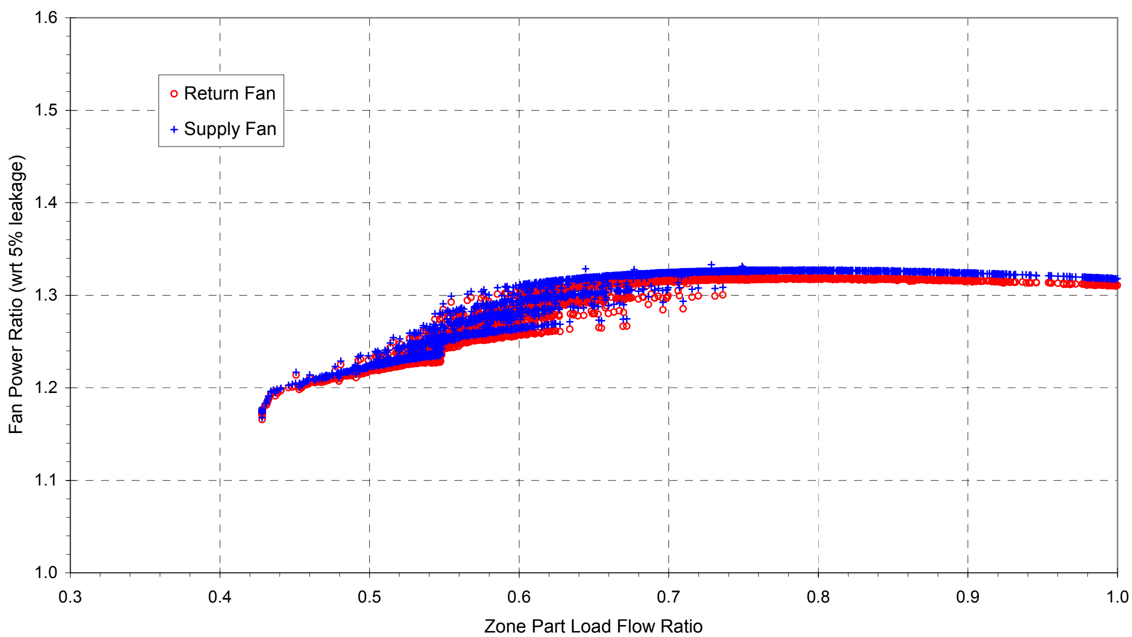

Figure 16. Upstream and Downstream Leaks (7.5+7.5) - Fan Power Impacts

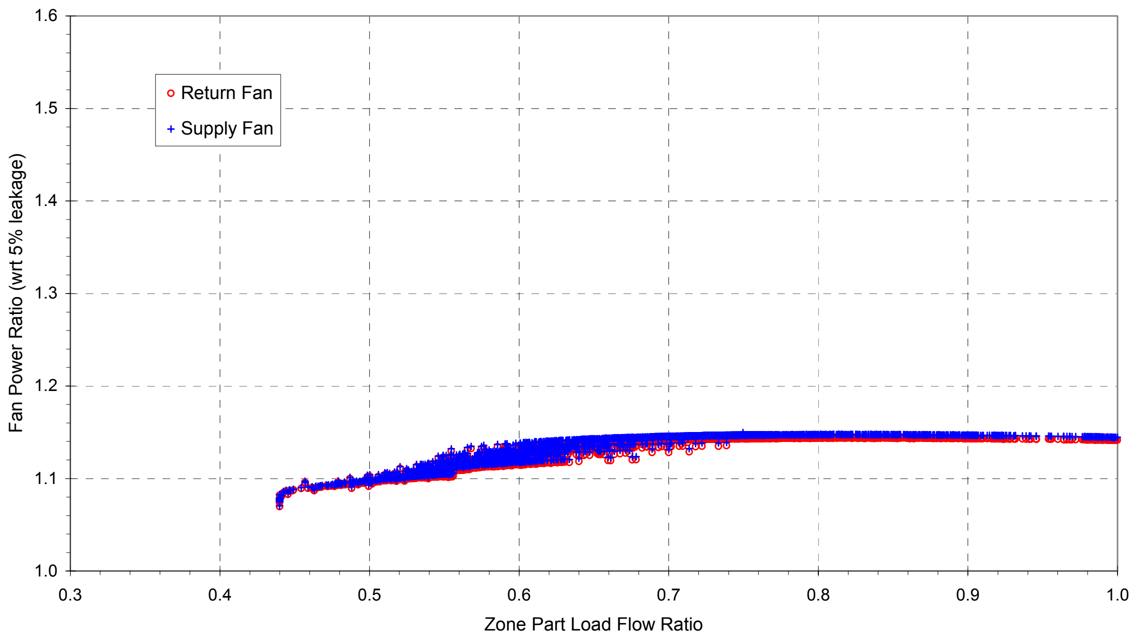

Figure 17. Upstream and Downstream Leaks (5+5) - Fan Power Impacts

6.2 Energy Consumption

Table III-1 in Appendix III summarizes the VAV distribution system energy performance for the 54 cases that we studied; the fractional energy uses by component and the energy increases due to duct leakage are listed in Tables III-2a and III-2b. The total HVAC site energy use reported includes supply and return fan electricity consumption, chiller and cooling tower electricity consumption, boiler electricity consumption, and boiler natural gas consumption. It does not include exhaust fan electricity consumption, which we did not model.

The coil loads listed in Table III-1 are large compared to total fan energy, but do not reflect end-use energy. However, the cooling and reheat coil loads can be related to plant site energy consumption by using the system-to-plant regression equations that we developed. Once this translation from coil loads to plant energy is made, Tables III-2a and III-2b show that annual total energy consumption for supply and return fans ranges from 10 to 25% of the total HVAC system energy consumption (17 to 33% of the total HVAC system electrical energy use). Annual cooling plant energy is the largest energy use component and ranges from 44 to 60% of the total HVAC system energy consumption (65 to 81% of the total HVAC system electrical energy use).

For comparison, California Energy Commission Year 2000 data (Brook 2002) indicate that about 36% of HVAC-related site electricity consumption in California's large commercial buildings is used by supply, return, and exhaust fans. Supply and return fans use about 60% of this fan-related energy, or about 22% of HVAC-related electricity consumption. This latter fraction is consistent with the midpoint of our range (17 to 33%). If we assume that the buildings that we simulated would use exhaust fan energy in the same proportion to supply and return fan energy as indicated by the CEC data, then our 17 to 33% supply and return fan energy fraction means that fans (supply, return, and exhaust) would use about 28 to 55% of HVAC-related electricity consumption.

Of particular interest are the fractional changes in site energy use resulting from duct leakage. These values are presented in the seven right-hand columns of Tables III-2a and III-2b. For 10+10 leakage, total fan energy increases by 40 to 50%. Cooling plant energy also increases (7 to 10%), but reheat energy decreases (3 to 10%). As described in the earlier discussion about dominant downstream leakage (Section 6.1), the reheat energy decreases with duct leakage due to VAV box operation at minimum turn down flows during reheating. In combination, the effect of fan and cooling energy increases (electrical), offset by reheat energy decreases (natural gas), increases total HVAC energy use by 2 to 14% for this case.

Compared to the significant increases in the 10+10 case, the increases are much smaller for 5+5 leakage: total fan energy increases by 10 to 14%, cooling plant energy increases by 2 to 3%, reheat energy decreases by 1 to 4%, and total HVAC energy increases by 0 to 4%.

In almost half the cases, the reheat energy decrease exceeds the corresponding cooling plant energy increase, particularly when downstream leakage is large. In a few cases, added duct leakage actually results in a slight reduction in total HVAC energy use compared to the tight duct case.

6.3 Equipment Sizing Considerations

An additional effect that duct leakage has on system performance is to increase the required size of system components. Tables IV-1a and IV-1b in Appendix IV summarize the maximum fan, VAV box, and zone airflows and the peak coil loads that occur over the annual simulation for the 54 cases that we analyzed. The impacts of duct leakage are presented in the four right-hand columns (fan airflows, VAV box airflows, and coil sizes), relative to the tight leakage case.

The fan size requirement increases by about 16 to 21% for 10+10 leakage. Both cooling and reheat coil size requirements increase: 7 to 12% for the cooling coil, and 2 to 6% for the reheat coils. Compared to the significant increases in the 10+10 case, the equipment size increases are much smaller for 5+5 leakage: 5 to 6% for the supply fan, 2 to 3% for the cooling coil, and 1 to 2% for the reheat coil.

The size increases (especially for the 10+10 case) are important because they translate into increased equipment capital costs, which are in addition to the increased energy operating costs described below.

6.4 HVAC System Operating Costs

Using our system-to-plant energy regression equations with energy cost data enables us to extend the simulation results to estimate duct leakage impacts on HVAC system operating costs. In particular, we calculated annual operating costs using year 2000 average commercial sector energy prices for California: \$0.0986/kWh and \$7.71/Million Btu (EIA 2003). These prices include demand charges, averaged over the total consumption for the year. In the discussion that follows, we ignored the separate effects of energy demand changes on demand charges. If demand charges were included, we expect that the actual operating cost increases would be larger than those reported here, because the largest fractional increases in energy use coincide with medium to full load operation of the HVAC system.

It is important to note that, because electrical energy costs much more (a factor of 3.7) than natural gas per unit of energy, the energy prices change the weighting of the energy contributions to the operating cost increases. Consequently, even the low total HVAC energy increases described in Section 6.2 still result in substantial cost increases in all but a few cases.

Tables V-1a and V-1b in Appendix V present our estimates of HVAC system annual operating costs for the various leakage cases, along with the changes in cost relative to the tight duct case. The combined chiller and cooling tower operating cost increase is equal to about half of the combined supply and return fan cost increase. Including the cost decreases associated with the heating plant, the total plant energy cost increase equals about one-third of the fan cost increase. For the 10+10 leakage case, HVAC system annual operating costs increase by 9 to 18% ($7,400 to $9,500) relative to the tight duct case. The increase for 5+5 leakage is 2 to 5% ($1,800 to $2,700).

The fractional and absolute cost increases do not necessarily correspond with each other, because the operating costs differ depending on building location and construction. For example, in the 10+10 leakage case, Tables V-1a and V-1b show that the 9% fractional cost increase is achieved by a $7,500 increase relative to an $87,000 "tight duct" operating cost ("new" Pasadena CZ9 building); the 18% fractional cost increase is achieved by an $8,200 increase relative to a $44,600 operating cost ("Title 24" Oakland CZ3 building). The $7,400 absolute cost increase is relative to a $63,300 operating cost ("Title 24" Pasadena CZ9 building) and corresponds to a fractional increase of about 12%; the $9,500 absolute increase is relative to a $77,300 operating cost ("old" Sacramento CZ12 building) and also corresponds to a fractional increase of about 12%.

6.5 Duct Sealing Cost Effectiveness

Figure 18 shows the range of increases in HVAC system annual operating costs due to leakage for all climates and building vintages, relative to the tight duct system (about 5% total leakage).

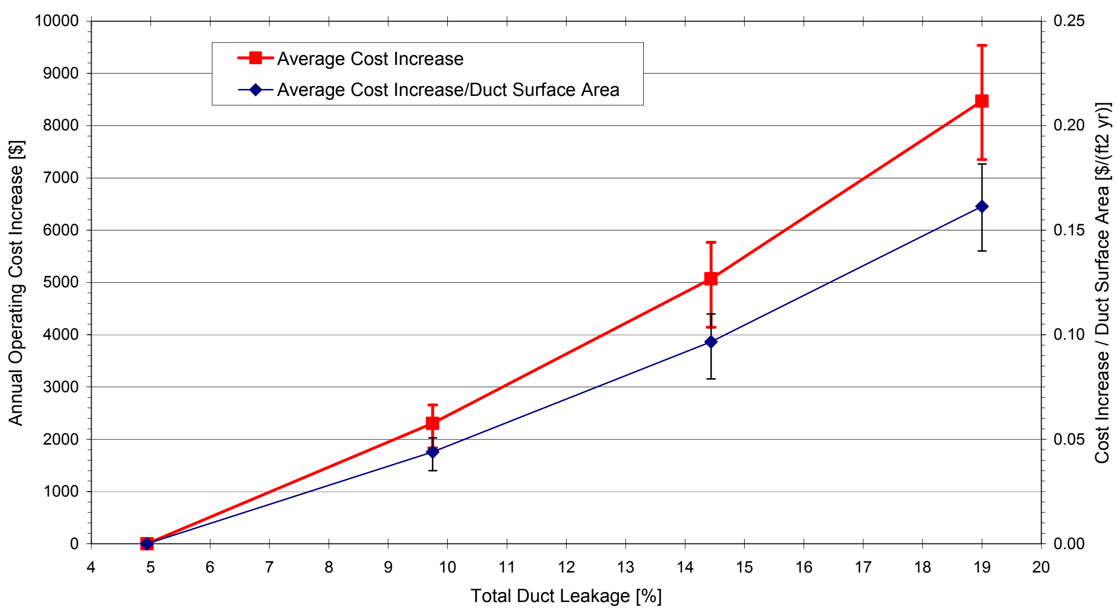

Figure 18. Duct Leakage Impacts on Annual HVAC Operating Costs

The values shown in Figure 18 assume that each floor's HVAC system, which serves 15,000 ft^2 of conditioned floor area, has a duct surface area of 5,250 ft^2. This surface area is based on commercial duct characterization data (Fisk et al. 2000). For large commercial HVAC systems, duct surface area ranges from 27 to 43% of the building floor area, and the area downstream of the VAV boxes ranges from 50 to 75% of the total duct surface area. Using typical ratios of 35% for the duct to floor area and 60% for the downstream duct area fraction, the duct surface area is 2,100 ft^2 upstream and 3,150 ft^2 downstream. Based on the total duct surface area (5,250 ft^2), the operating costs compared to the tight system increase by 0.03 to 0.05 $\$/ft^2$ for the 5+5 leakage case, 0.08 to 0.11 $\$/ft^2$ for the 7.5+7.5 case, and 0.14 to 0.18 $\$/ft^2$ for the 10+10 leakage case.

Duct sealing costs vary with fitting-to-straight-duct ratio, pressure class, and other system variables. Tsal et al. (1998) have suggested that an upper bound for the one-time cost of duct sealing is $\$0.25/ft^2$ of duct surface area. SMACNA has suggested that a reasonable average sealing cost is $\$0.20/ft^2$ for new commercial installations (Stratton 1998). SMACNA could not provide a sealing cost estimate for retrofitting existing systems due to wide cost variations resulting from system variables and sealing methods.

Assuming a one-time duct sealing cost of $\$0.20/ft^2$, the average simple payback for the duct sealing is 5 years for the 5+5 leakage case, 2 years for the 7.5+7.5 case, and 1.3 years for the 10+10 leakage case.

7. CONCLUSIONS

Our DOE-2/TRNSYS simulations indicate that a leaky VAV system (total leakage of about 19%) will use about 40 to 50% more fan energy annually than a tight system (about 5% leakage). Annual cooling plant energy also increases by about 7 to 10%, but reheat energy decreases (about 3 to 10%). In combination, the increase in total annual HVAC site energy is 2 to 14%. The total HVAC site energy use includes supply and return fan electricity consumption, chiller and cooling tower electricity consumption, boiler electricity consumption, and boiler natural gas consumption.

Using year 2000 average commercial sector energy prices for California ($0.0986/kWh and $7.71/Million Btu), the energy increases result in HVAC system annual operating cost increases ranging from 9 to 18% ($7,400 to $9,500). The low increases in total energy correspond to cases with large reductions in natural-gas-based reheat energy consumption due to the added leakage; the reheat reductions tend to offset the large electrical-based fan and cooling plant energy increases due to the added leakage. However, because electrical energy costs much more than natural gas per unit of energy, even the low total energy increases still result in substantial cost increases.

Normalized by duct surface area, the increases in HVAC system annual operating costs are 0.14 to 0.18 $\$/ft^2$ for the 19% leakage case. Using a suggested one-time duct sealing cost of $\$0.20/ft^2$ of duct surface area, these results indicate that sealing leaky ducts in VAV systems has a simple payback period of about 1.3 years. Even with total leakage rates as low as 10%, duct sealing is still cost effective. This suggests that duct sealing should be considered at least for VAV systems with 10% or more total duct leakage.

8. OTHER ISSUES AND IMPLICATIONS

Before duct performance in large commercial buildings can be accounted for in Title 24 nonresidential building energy standards, there are several issues that must be addressed and resolved. These include:

1. Specifying reliable duct air leakage measurement techniques that can be practically applied in the large commercial building sector.

2. Defining the duct leakage condition for the standard building used in Title 24 compliance simulations.
3. Assuring consistency between simulated duct performance impacts and actual impacts.
4. Developing compliance tests for the Alternative Calculation Method (ACM) Approval Manual (CEC 2001b) to evaluate duct performance simulations.

Regarding Issues 1 and 2, new duct air leakage measurement techniques for large commercial buildings are already under development at LBNL. These efforts are focused on developing a rapid technique that measures leakage flows rather than leakage area, and we expect that it could be used to populate a database of duct leakage conditions in the existing building stock.

After the "typical" duct leakage for the building stock is defined, then a decision can be made about what duct leakage level to assign to the standard building. If the standard building description includes a typical duct air leakage rate, then proposed buildings will be rewarded for sealing ducts. If instead the standard building has a reduced leakage level, proposed buildings that are not sealed will be penalized. The decision about what leakage level to assume for the standard building description will depend upon the preparedness of the market to handle required duct efficiency improvements, as opposed to optional improvements.

In terms of prescriptive compliance options, if the standard-building duct performance parameters are established to correspond to typical duct air leakage, determining compliance using the prescriptive approach is straightforward. If the proposed building has a typical duct air leakage level and has ducts insulated to Title 24 requirements, the building complies with respect to ducts. In other words with nothing done to improve duct performance in the building, it would meet the minimal duct performance level in this case. On the other hand, if the standard building has tighter-than-typical duct air leakage specifications, then compliance would require either performance measurements (i.e., duct air leakage measurements), or increased energy efficiency of other building components.

With the standard building defined as having leaky ducts, improving the duct performance in the proposed building affects compliance only if the performance budget approach is used. If leaks are sealed as a compliance conservation measure, standardized testing methods must be adopted for the verification of reduced leakage rates. Leakage rates determined from the tests would be part of the duct performance input data in the performance compliance analysis for the proposed building.

For Issue 3, one study has already shown through detailed minute-by-minute field measurements in a large commercial building that duct leakage has a significant impact on HVAC system performance (Diamond et al. 2003). The extensive set of HVAC system performance data collected by Diamond et al. could be used to validate simulation tools that are used to predict the duct performance impacts.

Regarding Issue 4, several tests must be performed already on alternative calculation methods before they are approved. Although a test does not yet exist, the proper modeling of duct performance in these alternative methods should be evaluated as part of these capability tests. Given that the current two certified nonresidential compliance tools depend upon DOE-2.1E as the reference evaluation program, and that DOE-2.1E cannot properly account for duct thermal performance, it is expected that results obtained using an alternative calculation method that properly accounts for duct thermal performance might differ substantially from the reference program results. Thus, we recommend that a new reference program be identified for use at least in this test (e.g., EnergyPlus). A prerequisite in this case is that the reference method be appropriately validated against field measurements.

Three additional steps will be required to further develop duct-modeling capabilities that address limitations in existing models and to initiate strong market activity related to duct system improvements. We recommend that these steps include:

1. Implementing duct models in user-friendly commercially-available software for building energy simulation, validating the implementations with case studies and demonstrations, and obtaining certification for software use as a primary or alternative compliance tool in support of the Title 24 Nonresidential Standards.
2. Developing methodologies to deal with airflows entering VAV boxes from ceiling return plenums (e.g., to model parallel fan-powered VAV boxes), to deal with duct surface heat transfer effects, and to deal with static pressure reset and supply air temperature reset strategies.
3. Transferring information to practitioners through publications, conferences, workshops, and other education programs.

GLOSSARY

ACM	Alternative Calculation Method
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers
CAV	Constant Air Volume
CEC	California Energy Commission
DOE	U.S. Department of Energy
EDS	Efficient Distribution Systems
EIA	Energy Information Administration
GWh	Giga Watt hours, 10^9 Wh, 10^6 kWh
HVAC	Heating, ventilating, and air conditioning
LBNL	Lawrence Berkeley National Laboratory
MW	Mega Watt, 10^6 W
PIER	Public Interest Energy Research
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
VAV	Variable Air Volume

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APPENDIX I: BUILDING SCHEDULES

Table I-1. Heating Set-Point Schedule (°F)

Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	5	60	1	5	60	1	5	60	1	5	60
6	7	65	6	16	65	6	16	65	6	16	65
8	18	70	17	24	60	17	24	60	17	24	60
19	19	65									
20	24	60									

Table I-2. Cooling Set-Point Schedule (°F)

Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	5	77	1	5	77	1	5	77	1	5	77
6	18	73	6	18	73	6	18	73	6	18	73
19	24	77	19	24	77	19	24	77	19	24	77

Table I-3. Lighting Schedule (Fraction of Full Intensity)

Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	4	5%	1	5	5%	1	5	5%	1	5	5%
5	5	10%	6	6	10%	6	7	10%	6	7	10%
6	6	20%	7	7	15%	8	17	15%	8	17	15%
7	7	40%	8	14	25%	18	20	10%	18	20	10%
8	8	70%	15	17	20%	21	24	5%	21	24	5%
9	9	80%	18	18	15%						
10	17	85%	19	24	10%						
18	18	80%									
19	19	35%									
20	24	10%									

Table I-4. Equipment Heat Gain Schedule (Fraction of Full Load)

Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	5	15%	1	7	15%	1	7	15%	1	7	15%
6	6	20%	8	8	20%	8	17	20%	8	17	20%
7	7	35%	9	14	25%	18	24	15%	18	24	15%
8	8	60%	15	17	20%						
9	16	70%	18	24	15%						
17	17	65%									
18	18	45%									
19	19	30%									
20	21	20%									
22	24	15%									

Table I-5. Air-Handler Operating Schedule (Supply and Return Fans)

HVAC Fan (On/Off)											
Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	5	Off	1	5	Off	1	24	Off	1	24	Off
6	20	On	6	15	On						
21	24	Off	16	24	Off						

Table I-6. Air Infiltration Schedule (Fraction of Full Infiltration Airflow)

Infiltration (%)											
Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	5	100%	1	5	100%	1	24	100%	1	24	100%
6	20	0%	6	15	0%						
21	24	100%	16	24	100%						

Table I-7. Occupancy Schedule (Fraction of Full Occupancy)

People (%)											
Weekday			Saturday			Sunday			Holiday		
Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value	Start (Hr)	Stop (Hr)	Value
1	4	0%	1	6	0%	1	7	0%	1	7	0%
5	5	5%	7	7	5%	8	20	5%	8	20	5%
6	6	10%	8	17	15%	21	24	0%	21	24	0%
7	7	25%	18	20	5%						
8	11	65%	21	24	0%						
12	13	60%									
14	17	65%									
18	18	40%									
19	19	25%									
20	20	10%									
21	24	5%									

APPENDIX II: REGRESSION EQUATIONS AND COEFFICIENTS

This Appendix lists the plant energy regression equations and their coefficients that we developed to translate the intermediate floor cooling and heating coil loads (predicted by the TRNSYS air-handling system simulations) to plant energy consumption and demand (i.e., chiller electricity, cooling tower electricity, boiler electricity, and boiler fuel).

Each equation correlates the energy demand predicted by DOE-2 in a given hour to the intermediate floor part load factor for that hour (PLRC for cooling, PLRH for heating). The part load factor is defined as the hourly coil load (summed over all zones on a floor, or over all zones in the building) divided by the maximum hourly coil load over all operating hours (summed over the corresponding floor or the entire building respectively).

Tables II-1a through II-2b demonstrate that the part load ratio for a single intermediate floor can be used to represent the part load ratio for the entire building. Therefore, we used the PLR's based on a single mid-height intermediate floor to translate the coil loads to plant energy consumption and demand.

Table II-1a. Cooling Part Load Ratio Equation (PLRC)

PLRC(Building) = A x PLRC(Intermediate Floor)

R^2 Range: 1.000

Climate Zone	Building	A
CZ3 (Oakland)	1980 1990 2005	0.994 0.995 0.995
CZ9 (Pasadena)	1980 1990 2005	1.005 1.002 0.997
CZ12 (Sacramento)	1980 1990 2005	0.996 0.996 0.997

Table II-1b. Cooling Part Load Ratio - Example Values

Cooling Part Load Ratio (Intermediate Floor)	Predicted Cooling Part Load Ratio (Whole Building)
0	0
0.1	0.099
0.5	0.497

Table II-2a. Heating Part Load Ratio Equation (PLRH)

PLRH(Building) = A x PLRH[Intermediate Floor] + B x PLRH[Intermediate Floor]^2

R^2 Range: 0.9999 to 1.000

Climate Zone	Building	A	B
CZ3 (Oakland)	1980	0.984	0.020
	1990	0.981	0.020
	2005	0.980	0.026
CZ9 (Pasadena)	1980	0.964	0.037
	1990	0.964	0.036
	2005	0.953	0.046
CZ12 (Sacramento)	1980	0.978	0.026
	1990	0.977	0.026
	2005	0.971	0.035

Table II-2b. Heating Part Load Ratio - Example Values

Heating Part Load Ratio (Intermediate Floor)	Predicted Heating Part Load Ratio (Whole Building)
0	0
0.1	0.099
0.5	0.497

Each cooling equation that follows in Tables II-3a, II-3b, and II-4a represents the dimensional function f(PLR) that is used in the following relation:

$$\text{Hourly Energy Demand} = (\text{TRNSYS Load}_{max,\ 10+10})/(\text{DOE-2 Load}_{max}) * f(\text{PLRC})$$

where

"TRNSYS $\text{Load}_{max,\ 10+10}$" is the *maximum* hourly cooling coil total load (sensible plus latent) determined using TRNSYS for the selected intermediate floor over all operating hours in the simulation case for the specified climate and building vintage combination, for the case with the maximum duct leakage (which requires the largest fans and coils), and

"DOE-2 Load_{max}" is the *maximum* hourly cooling coil total load determined using DOE-2 for the same floor, climate, and building vintage case.

The ratio of the TRNSYS and DOE-2 coil loads serves as a correction to account for different equipment sizes. Specifically, we assume that plant size scales linearly with coil size. This means that an air-handling system with duct leakage (simulated by TRNSYS) that uses a cooling coil 50% larger than the one used in the associated DOE-2 simulation (with no duct leakage) will result in 50% more chiller and cooling tower electricity being consumed at a given part load.

In the equation above, the parameter PLRC is the *hourly* coil part load factor determined using the hourly and maximum cooling coil loads from TRNSYS for the same floor, climate, and building vintage case:

$$PLRC_{hour} = TRNSYS\ Load_{hour} / TRNSYS\ Load_{max,\ 10+10}$$

The $PLRC_{hour}$ relation assumes that the same size fans and coils are used within a given climate and building vintage set of cases regardless of the leakage condition, and that the equipment sizes in those cases are based on the sizes required to meet the loads in the maximum leakage condition.

A similar set of relations is used with the heating equations in Tables II-5a, II-5b, II-6a, and II-6b, which depend on PLRH rather than PLRC.

Table II-3a. Chiller Electricity Equation - Low PLRC

(PLRC[Intermediate Floor] <=~0.09)

Chiller Electricity (Building, kW) = A + B x PLRC[Intermediate Floor]

R^2 Range: 0.9999 to 1.0000

Climate Zone	Building	PLRC<=	A	B
CZ3 (Oakland)	1980	0.089	9.185	426.201
	1990	0.089	8.938	415.787
	2005	0.091	7.310	335.231
CZ9 (Pasadena)	1980	0.086	12.395	602.760
	1990	0.091	12.088	549.800
	2005	0.097	10.283	438.590
CZ12 (Sacramento)	1980	0.093	13.158	592.039
	1990	0.093	12.684	566.690
	2005	0.095	9.890	435.552

Table II-3b. Chiller Electricity Equation - High PLRC

(PLRC[Intermediate Floor] >~ 0.09)

Chiller Electricity (Building, kW) =
A + B x PLRC[Intermediate Floor] + C x PLRC[Intermediate Floor]^2

R^2 Range: 0.9997 to 0.9998

Climate Zone	Building	PLRC>	A	B	C
CZ3 (Oakland)	1980	0.089	41.674	54.981	65.847
	1990	0.089	40.672	53.564	63.780
	2005	0.091	33.277	43.117	50.759
CZ9 (Pasadena)	1980	0.086	56.763	74.223	101.610
	1990	0.091	55.621	66.529	87.714
	2005	0.097	47.259	52.271	68.044
CZ12 (Sacramento)	1980	0.093	60.760	69.440	97.390
	1990	0.093	58.561	66.482	92.559
	2005	0.095	45.499	51.773	69.780

Table II-3c. Chiller Electricity Consumption - Example Values

Cooling Part Load Ratio (Intermediate Floor)	Predicted Chiller Electricity Consumption (kW) (Whole Building)
0	0
0.08	43.3
0.50	85.6

Table II-4a. Cooling Tower Electricity Equation

Cooling Tower Electricity (Building, kW) =
A + B x PLRC[Intermediate Floor] + C x PLRC[Intermediate Floor]^2 +
D x PLRC[Intermediate Floor]^3 + E x PLRC[Intermediate Floor]^4

R^2 Range: 0.9338 to 0.9997

Climate Zone	Building	A	B	C	D	E
CZ3 (Oakland)	1980	14.614	1.591	7.699	-9.158	4.965
	1990	14.267	1.514	7.547	-9.048	4.907
	2005	11.669	1.252	6.139	-7.374	4.011
CZ9 (Pasadena)	1980	19.859	0.208	27.112	-62.199	51.365
	1990	19.390	0.296	22.953	-50.140	38.975
	2005	16.325	3.979	-2.383	0.365	6.147
CZ12 (Sacramento)	1980	20.955	2.906	7.532	-15.125	15.643
	1990	20.198	2.783	7.468	-15.694	15.968
	2005	15.724	3.131	0.236	-0.982	5.966

Table II-4b. Cooling Tower Electricity Consumption - Example Values

Cooling Part Load Ratio (Intermediate Floor)	Predicted Cooling Tower Electricity Consumption (kW) (Whole Building)
0	0.000
0.5	16.500
0.8	18.159

Table II-5a. Boiler Electricity Equation - Low PLRH

(PLRH[Intermediate Floor] <= ~ 0.40)

Boiler Electricity (Building, kW) =
A + B x PLRH[Intermediate Floor] + C x PLRH[Intermediate Floor]^2

R^2 Range: 0.9924 to 0.9977

Climate Zone	Building	PLRH <=	A	B	C
CZ3 (Oakland)	1980	0.387	0.530	57.568	2.401
	1990	0.370	0.495	56.688	2.303
	2005	0.410	0.402	41.619	1.344
CZ9 (Pasadena)	1980	0.310	0.392	54.011	7.356
	1990	0.315	0.371	52.832	4.766
	2005	0.348	0.322	34.293	4.139
CZ12 (Sacramento)	1980	0.357	0.661	74.052	8.906
	1990	0.335	0.583	72.013	5.284
	2005	0.345	0.380	43.743	5.713

Table II-5b. Boiler Electricity Equation - High PLRH

(PLRH[Intermediate Floor] > ~ 0.40)

Boiler Electricity (Building, kW) = A

R^2 Range: 1.000

Climate Zone	Building	PLRH >	A
CZ3 (Oakland)	1980	0.387	23.195
	1990	0.370	21.731
	2005	0.410	17.797
CZ9 (Pasadena)	1980	0.310	17.966
	1990	0.315	17.478
	2005	0.348	12.747
CZ12 (Sacramento)	1980	0.357	28.270
	1990	0.335	25.661
	2005	0.345	16.149

Table II-5c. Boiler Electricity Consumption - Example Values

Heating Part Load Ratio (Intermediate Floor)	Predicted Boiler Electricity Consumption (kW) (Whole Building)
0	0.0
0.2	12.1
0.5	23.2

Table II-6a. Boiler Fuel Equation - Low PLRH

(PLRH[Intermediate Floor] <= ~ 0.40)

Boiler Fuel (Building, Btu/h) =
$A + B \times PLRH[\text{Intermediate Floor}] + C \times PLRH[\text{Intermediate Floor}]^2$

R^2 Range: 0.9990 to 0.9997

Climate Zone	Building	PLRH <=	A	B	C
CZ3 (Oakland)	1980	0.387	33,611	3,648,544	152,214
	1990	0.370	31,341	3,592,824	145,799
	2005	0.410	25,463	2,637,797	84,964
CZ9 (Pasadena)	1980	0.310	24,834	3,423,115	466,319
	1990	0.315	23,536	3,348,410	301,913
	2005	0.348	20,388	2,173,399	262,440
CZ12 (Sacramento)	1980	0.357	41,880	4,693,275	564,484
	1990	0.335	37,007	4,563,984	335,031
	2005	0.345	24,089	2,772,405	361,904

Table II-6b. Boiler Fuel Equation - High PLRH

(PLRH[Intermediate Floor] > ~ 0.40)

Boiler Fuel (Building, Btu/h) = $A + B \times PLRH[\text{Intermediate Floor}]$

R^2 Range: 0.9984 to 0.9996

Climate Zone	Building	PLRH >	A	B
CZ3 (Oakland)	1980	0.387	447,637	2,651,439
	1990	0.370	423,385	2,603,255
	2005	0.410	333,343	1,934,786
CZ9 (Pasadena)	1980	0.310	365,870	2,538,821
	1990	0.315	356,207	2,452,285
	2005	0.348	244,708	1,646,832
CZ12 (Sacramento)	1980	0.357	562,719	3,462,940
	1990	0.335	516,927	3,318,081
	2005	0.345	319,932	2,068,909

Table II-6c. Boiler Fuel Consumption - Example Values

Heating Part Load Ratio (Intermediate Floor)	Predicted Boiler Fuel Consumption (Btu/h) (Whole Building)
0	0
0.2	769,400
0.5	1,773,400

APPENDIX III: ENERGY PERFORMANCE IMPACTS

Table III-1. Leakage Impacts on Annual Energy Use

		Annual Site Energy Use [MWh]												
CZ	Vintage	Supply Fan	Return Fan	Both Fans	Cooling Coil	Reheat Coils	Chiller Electricity	Tower Electricity	Cooling Energy	Boiler Electricity	Boiler Fuel	Heating Energy	Total Electricity	Total Energy
		10 + 10 (Upstream + Downstream) Leaks												
3	Old	141	49	190	1,201	232	297	77	374	13	257	270	578	835
	New	141	49	190	1,194	204	292	75	367	12	236	249	570	806
	T24	121	43	164	1,041	146	255	67	322	8	154	162	494	648
9	Old	134	47	181	2,572	323	531	104	635	19	406	425	835	1,241
	New	133	47	179	2,626	421	513	103	616	24	524	548	819	1,343
	T24	109	39	148	2,120	175	415	91	505	9	200	209	663	862
12	Old	154	54	208	2,162	385	439	94	532	23	441	464	763	1,204
	New	152	53	205	2,101	332	423	90	513	20	400	420	739	1,139
	T24	118	42	160	1,737	184	350	77	427	11	215	226	597	813
		7.5 + 7.5 (Upstream + Downstream) Leaks												
3	Old	122	43	165	1,109	235	285	76	361	14	261	275	540	801
	New	122	43	165	1,107	208	281	75	355	13	241	254	532	773
	T24	105	37	142	959	148	245	66	311	8	156	164	461	616
9	Old	117	41	159	2,466	334	516	103	618	19	421	440	796	1,217
	New	117	41	158	2,495	437	495	102	597	25	543	568	780	1,323
	T24	95	34	129	2,026	180	403	90	493	10	206	216	632	837
12	Old	134	47	181	2,077	389	427	93	520	23	446	469	724	1,169
	New	132	46	178	2,019	336	412	90	502	21	405	425	701	1,105
	T24	102	36	139	1,667	186	340	76	417	11	218	229	566	784
		10 + 2.5 (Upstream + Downstream) Leaks												
3	Old	120	42	162	1,096	242	283	76	359	14	269	283	535	804
	New	119	42	161	1,091	216	279	75	353	13	250	263	528	778
	T24	102	36	138	943	152	243	66	309	8	160	168	455	615
9	Old	120	42	162	2,492	359	520	103	623	21	452	473	805	1,257
	New	121	43	164	2,534	470	501	102	603	27	583	610	793	1,376
	T24	95	34	129	2,031	191	404	90	494	10	219	229	633	852
12	Old	132	47	179	2,070	396	426	93	518	24	454	478	721	1,176
	New	131	46	177	2,010	344	410	90	500	21	415	436	698	1,113
	T24	101	36	136	1,656	190	339	76	415	11	224	235	563	787
		2.5 + 10 (Upstream + Downstream) Leaks												
3	Old	110	39	148	1,044	232	276	76	352	13	257	270	513	770
	New	109	38	148	1,042	204	272	74	346	12	236	249	506	743
	T24	94	33	127	904	146	237	66	303	8	154	162	438	592
9	Old	102	36	138	2,347	323	498	101	599	19	406	425	756	1,162
	New	102	36	137	2,365	421	478	101	579	24	524	548	740	1,264
	T24	84	30	113	1,940	175	392	89	482	9	200	209	604	804
12	Old	119	42	161	2,010	385	417	93	510	23	441	464	693	1,135
	New	117	41	159	1,955	332	403	90	492	20	400	420	671	1,071
	T24	91	32	124	1,615	184	333	76	409	11	215	226	543	759
		5 + 5 (Upstream + Downstream) Leaks												
3	Old	107	38	144	1,028	239	274	76	350	14	265	279	508	773
	New	107	38	144	1,026	212	270	74	344	13	246	259	501	747
	T24	91	32	124	885	150	235	66	301	8	158	166	433	591
9	Old	104	37	141	2,369	346	501	101	603	20	436	456	764	1,200
	New	105	37	142	2,401	453	483	101	584	26	563	589	751	1,314
	T24	84	30	113	1,942	185	392	89	482	10	212	222	605	817
12	Old	117	41	158	2,001	392	416	93	509	23	450	473	691	1,141
	New	116	41	156	1,945	340	401	89	491	21	410	431	668	1,078
	T24	89	32	121	1,603	188	332	76	408	11	221	232	540	761
		2.5 + 2.5 (Upstream + Downstream) Leaks												
3	Old	94	33	128	953	242	264	76	339	14	269	283	481	750
	New	94	33	127	952	216	260	74	334	13	250	264	474	725
	T24	80	29	109	820	152	226	66	292	8	160	168	409	569
9	Old	93	33	126	2,282	359	489	101	589	21	452	473	737	1,189
	New	95	34	128	2,314	470	472	101	572	27	583	610	727	1,310
	T24	74	26	101	1,866	191	383	89	472	10	219	229	583	802
12	Old	104	37	140	1,933	396	406	92	499	24	455	478	663	1,117
	New	102	36	139	1,878	344	392	89	481	21	415	437	641	1,056
	T24	79	28	107	1,546	190	324	76	400	11	224	235	518	742

Table III-2a. Fractional Impacts of Leakage on Energy Uses

		Total Energy Use [%]			Electrical Use [%]			Energy Increase Due to Leakage [%]						
CZ	Vintage	Both Fans	Cooling Energy	Heating Energy	Both Fans	Cooling Energy	Heating Energy	Both Fans	Cooling Coil	Reheat Coils	Cooling Energy	Heating Energy	Total Electricity	Total Energy
		10 + 10 (Upstream + Downstream) Leaks												
3	Old	23	45	32	33	65	2	49	26	-4	10	-5	20	11
	New	24	46	31	33	65	2	49	25	-5	10	-6	20	11
	T24	25	50	25	33	65	2	50	27	-4	10	-4	21	14
9	Old	15	51	34	22	76	2	43	13	-10	8	-10	13	4
	New	13	46	41	22	75	3	40	13	-10	8	-10	13	2
	T24	17	59	24	22	76	1	47	14	-9	7	-9	14	8
12	Old	17	44	39	27	70	3	48	12	-3	7	-3	15	8
	New	18	45	37	28	69	3	48	12	-3	7	-4	15	8
	T24	20	53	28	27	71	2	50	12	-3	7	-4	15	10
	Avg	19	49	32	27	70	2	47	17	-6	8	-6	16	8
	Min	13	44	24	22	65	1	40	12	-10	7	-10	13	2
	Max	25	59	41	33	76	3	50	27	-3	10	-3	21	14
		7.5 + 7.5 (Upstream + Downstream) Leaks												
3	Old	21	45	34	31	67	3	29	16	-3	6	-3	12	7
	New	21	46	33	31	67	2	29	16	-4	6	-4	12	7
	T24	23	50	27	31	67	2	30	17	-2	6	-3	13	8
9	Old	13	51	36	20	78	2	25	8	-7	5	-7	8	2
	New	12	45	43	20	77	3	23	8	-7	4	-7	7	1
	T24	15	59	26	20	78	2	28	9	-6	4	-6	8	4
12	Old	15	44	40	25	72	3	29	7	-2	4	-2	9	5
	New	16	45	38	25	72	3	29	7	-2	4	-3	9	5
	T24	18	53	29	24	74	2	30	8	-2	4	-3	9	6
	Avg	17	49	34	25	72	2	28	11	-4	5	-4	10	5
	Min	12	44	26	20	67	2	23	7	-7	4	-7	7	1
	Max	23	59	43	31	78	3	30	17	-2	6	-2	13	8
		10 + 2.5 (Upstream + Downstream) Leaks												
3	Old	20	45	35	30	67	3	27	15	0	6	0	11	7
	New	21	45	34	31	67	2	27	15	0	6	0	11	7
	T24	22	50	27	30	68	2	27	15	0	6	0	11	8
9	Old	13	50	38	20	77	3	28	9	0	6	0	9	6
	New	12	44	44	21	76	3	28	10	0	5	0	9	5
	T24	15	58	27	20	78	2	28	9	0	5	0	9	6
12	Old	15	44	41	25	72	3	28	7	0	4	0	9	5
	New	16	45	39	25	72	3	27	7	0	4	0	9	5
	T24	17	53	30	24	74	2	28	7	0	4	0	9	6
	Avg	17	48	35	25	72	3	27	10	0	5	0	10	6
	Min	12	44	27	20	67	2	27	7	0	4	0	9	5
	Max	22	58	44	31	78	3	28	15	0	6	0	11	8

Table III-2b. Fractional Impacts of Leakage on Energy Uses

		Total Energy Use [%]			Electrical Use [%]			Energy Increase Due to Leakage [%]						
CZ	Vintage	Both Fans	Cooling Energy	Heating Energy	Both Fans	Cooling Energy	Heating Energy	Both Fans	Cooling Coil	Reheat Coils	Cooling Energy	Heating Energy	Total Electricity	Total Energy
		2.5 + 10 (Upstream + Downstream) Leaks												
3	Old	19	46	35	29	69	3	16	10	-4	4	-5	7	3
	New	20	47	34	29	68	2	16	9	-5	4	-6	7	3
	T24	22	51	27	29	69	2	17	10	-4	4	-4	7	4
9	Old	12	52	37	18	79	2	9	3	-10	2	-10	3	-2
	New	11	46	43	19	78	3	7	2	-10	1	-10	2	-4
	T24	14	60	26	19	80	2	12	4	-9	2	-9	4	0
12	Old	14	45	41	23	74	3	14	4	-3	2	-3	5	2
	New	15	46	39	24	73	3	15	4	-3	2	-4	5	1
	T24	16	54	30	23	75	2	16	4	-3	2	-4	5	2
	Avg	16	50	35	24	74	2	14	6	-6	3	-6	5	1
	Min	11	45	26	18	68	2	7	2	-10	1	-10	2	-4
	Max	22	60	43	29	80	3	17	10	-3	4	-3	7	4
		5 + 5 (Upstream + Downstream) Leaks												
3	Old	19	45	36	28	69	3	13	8	-1	3	-2	6	3
	New	19	46	35	29	69	3	13	8	-2	3	-2	6	3
	T24	21	51	28	29	70	2	14	8	-1	3	-1	6	4
9	Old	12	50	38	18	79	3	11	4	-3	2	-4	4	1
	New	11	44	45	19	78	3	10	4	-4	2	-3	3	0
	T24	14	59	27	19	80	2	12	4	-3	2	-3	4	2
12	Old	14	45	41	23	74	3	13	4	-1	2	-1	4	2
	New	15	46	40	23	73	3	13	4	-1	2	-1	4	2
	T24	16	54	30	22	76	2	13	4	-1	2	-1	4	3
	Avg	16	49	36	23	74	3	13	5	-2	2	-2	5	2
	Min	11	44	27	18	69	2	10	4	-4	2	-4	3	0
	Max	21	59	45	29	80	3	14	8	-1	3	-1	6	4
		2.5 + 2.5 (Upstream + Downstream) Leaks												
3	Old	17	45	38	27	71	3							
	New	18	46	36	27	70	3							
	T24	19	51	30	27	71	2							
9	Old	11	50	40	17	80	3							
	New	10	44	47	18	79	4							
	T24	13	59	29	17	81	2							
12	Old	13	45	43	21	75	4							
	New	13	46	41	22	75	3							
	T24	14	54	32	21	77	2							
	Avg	14	49	37	22	75	3							
	Min	10	44	29	17	70	2							
	Max	19	59	47	27	81	4							

APPENDIX IV: EQUIPMENT SIZING IMPACTS

Table IV-1a. Leakage Impacts on Equipment Sizing

		Maximum Airflows [scfm]			Zone Part Load Flow Ratio		Maximum Load [kW]		Increase Due to Leakage [%]			
CZ	Vintage	Supply Fan	VAV Boxes	Zone Supply	Avg	RMS	Cooling Coil	Reheat Coils	Supply Fan	VAV Boxes	Cooling Coil	Reheat Coils
		10 + 10 (Upstream + Downstream) Leaks										
	Old	14,615	13,311	11,980	0.57	0.13	138	77	16	8	8	5
3	New	14,513	13,212	11,891	0.57	0.14	134	69	16	8	8	5
	T24	11,317	10,264	9,238	0.62	0.13	119	59	17	8	8	6
	Old	18,078	15,956	14,361	0.59	0.09	155	52	19	8	7	2
9	New	17,988	15,612	14,051	0.63	0.07	160	54	21	8	12	2
	T24	12,756	11,235	10,111	0.65	0.09	141	38	19	8	9	4
	Old	17,463	15,774	14,197	0.58	0.12	175	99	17	8	8	6
12	New	17,001	15,363	13,826	0.58	0.13	169	86	17	8	8	5
	T24	11,719	10,503	9,453	0.65	0.12	142	53	18	8	9	5
				Avg	0.61	0.11			18	8	9	4
				Min	0.57	0.07			16	8	7	2
				Max	0.65	0.14			21	8	12	6
		7.5 + 7.5 (Upstream + Downstream) Leaks										
	Old	13,877	12,951	11,980	0.57	0.13	134	75	10	5	5	3
3	New	13,779	12,855	11,891	0.58	0.13	131	68	10	5	5	3
	T24	10,735	9,987	9,238	0.62	0.13	116	58	11	5	5	4
	Old	17,031	15,525	14,361	0.60	0.08	151	52	12	5	5	1
9	New	16,877	15,190	14,051	0.64	0.07	149	54	13	5	5	1
	T24	12,011	10,931	10,111	0.66	0.09	137	37	12	5	6	2
	Old	16,547	15,348	14,197	0.58	0.12	170	97	11	5	5	4
12	New	16,110	14,947	13,826	0.58	0.13	164	85	11	5	5	3
	T24	11,083	10,219	9,453	0.65	0.12	137	52	11	5	6	3
				Avg	0.61	0.11			11	5	5	3
				Min	0.57	0.07			10	5	5	1
				Max	0.66	0.13			13	5	6	4
		10 + 2.5 (Upstream + Downstream) Leaks										
	Old	13,491	12,287	11,980	0.58	0.12	133	73	7	0	4	0
3	New	13,396	12,196	11,891	0.58	0.13	130	66	7	0	4	0
	T24	10,447	9,475	9,238	0.63	0.12	115	56	8	0	4	0
	Old	16,687	14,729	14,361	0.61	0.08	151	51	10	0	5	0
9	New	16,605	14,411	14,051	0.66	0.06	152	53	11	0	7	0
	T24	11,775	10,370	10,111	0.67	0.08	137	36	10	0	5	0
	Old	16,120	14,561	14,197	0.59	0.11	169	93	8	0	4	0
12	New	15,693	14,181	13,826	0.59	0.12	163	82	8	0	4	0
	T24	10,818	9,695	9,453	0.66	0.11	136	51	9	0	5	0
				Avg	0.62	0.10			9	0	5	0
				Min	0.58	0.06			7	0	4	0
				Max	0.67	0.13			11	0	7	0

Table IV-1b. Leakage Impacts on Equipment Sizing

		Maximum Airflows [scfm]			Zone Part Load Flow Ratio		Maximum Load [kW]		Increase Due to Leakage [%]			
CZ	Vintage	Supply Fan	VAV Boxes	Zone Supply	Avg	RMS	Cooling Coil	Reheat Coils	Supply Fan	VAV Boxes	Cooling Coil	Reheat Coils
		2.5 + 10 (Upstream + Downstream) Leaks										
3	Old	13,612	13,311	11,980	0.57	0.13	132	77	8	8	3	5
	New	13,512	13,212	11,891	0.57	0.14	129	69	8	8	3	5
	T24	10,507	10,264	9,238	0.62	0.13	114	59	8	8	4	6
9	Old	16,446	15,956	14,361	0.59	0.09	148	52	8	8	3	2
	New	16,160	15,612	14,051	0.63	0.07	143	54	8	8	1	2
	T24	11,586	11,235	10,111	0.65	0.09	134	38	8	8	3	4
12	Old	16,164	15,774	14,197	0.58	0.12	167	99	8	8	4	6
	New	15,741	15,363	13,826	0.58	0.13	161	86	8	8	3	5
	T24	10,784	10,503	9,453	0.65	0.12	134	53	8	8	4	5
				Avg	0.61	0.11			8	8	3	4
				Min	0.57	0.07			8	8	1	2
				Max	0.65	0.14			8	8	4	6
		5 + 5 (Upstream + Downstream) Leaks										
3	Old	13,195	12,610	11,980	0.58	0.13	131	74	5	3	2	1
	New	13,101	12,517	11,891	0.58	0.13	127	67	5	3	2	1
	T24	10,197	9,724	9,238	0.62	0.13	113	57	5	3	3	2
9	Old	16,068	15,116	14,361	0.61	0.08	147	51	6	3	2	1
	New	15,857	14,790	14,051	0.65	0.06	145	54	6	3	2	1
	T24	11,326	10,643	10,111	0.66	0.09	133	37	6	3	3	1
12	Old	15,702	14,944	14,197	0.58	0.12	166	95	5	3	3	2
	New	15,289	14,554	13,826	0.59	0.12	160	83	5	3	3	2
	T24	10,496	9,951	9,453	0.66	0.11	133	51	5	3	3	1
				Avg	0.61	0.11			5	3	2	1
				Min	0.58	0.06			5	3	2	1
				Max	0.66	0.13			6	3	3	2
		2.5 + 2.5 (Upstream + Downstream) Leaks										
3	Old	12,565	12,287	11,980	0.58	0.12	128	73				
	New	12,473	12,196	11,891	0.58	0.13	125	66				
	T24	9,699	9,475	9,238	0.63	0.12	110	56				
9	Old	15,181	14,729	14,361	0.61	0.08	144	51				
	New	14,917	14,411	14,051	0.66	0.06	142	53				
	T24	10,695	10,370	10,111	0.67	0.08	130	36				
12	Old	14,921	14,561	14,197	0.59	0.11	162	94				
	New	14,530	14,181	13,826	0.59	0.12	156	82				
	T24	9,954	9,695	9,453	0.66	0.11	130	51				
				Avg	0.62	0.10						
				Min	0.58	0.06						
				Max	0.67	0.13						

APPENDIX V: OPERATING COST IMPACTS

Table V-1a. Leakage Impacts on Annual HVAC System Operating Costs

		Annual Operating Cost [$]				Cost Increase Due to Leakage					
		Both				Fans	Cooling	Heating	Total		
CZ	Vintage	Fans	Cooling	Heating	Total	$/yr	$/yr	$/yr	$/yr	%	$/ft2
		10 + 10 (Upstream + Downstream) Leaks									
3	Old	18,754	36,907	8,088	63,749	6,170	3,457	-390	9,237	17	0.18
	New	18,721	36,227	7,435	62,383	6,172	3,320	-445	9,047	17	0.17
	T24	16,134	31,793	4,842	52,769	5,409	2,994	-198	8,204	18	0.16
9	Old	17,848	62,643	12,523	93,014	5,376	4,533	-1,422	8,487	10	0.16
	New	17,694	60,689	16,127	94,510	5,049	4,264	-1,837	7,475	9	0.14
	T24	14,598	49,836	6,178	70,611	4,658	3,299	-605	7,352	12	0.14
12	Old	20,489	52,490	13,851	86,831	6,638	3,334	-435	9,538	12	0.18
	New	20,254	50,630	12,508	83,392	6,592	3,163	-506	9,249	12	0.18
	T24	15,782	42,077	6,715	64,574	5,249	2,675	-283	7,642	13	0.15
				Avg	74,648	5,701	3,449	-680	8,470	13	0.16
				Min	52,769	4,658	2,675	-1,837	7,352	9	0.14
				Max	94,510	6,638	4,533	-198	9,538	18	0.18
		7.5 + 7.5 (Upstream + Downstream) Leaks									
3	Old	16,270	35,594	8,217	60,081	3,685	2,145	-261	5,569	10	0.11
	New	16,235	35,014	7,583	58,832	3,686	2,107	-298	5,496	10	0.10
	T24	13,958	30,662	4,908	49,528	3,233	1,862	-132	4,963	11	0.09
9	Old	15,631	60,971	12,974	89,576	3,159	2,861	-970	5,050	6	0.10
	New	15,602	58,853	16,724	91,178	2,957	2,427	-1,240	4,144	5	0.08
	T24	12,689	48,618	6,378	67,685	2,750	2,081	-405	4,426	7	0.08
12	Old	17,803	51,262	13,997	83,062	3,952	2,106	-289	5,769	7	0.11
	New	17,588	49,465	12,674	79,726	3,925	1,999	-340	5,583	8	0.11
	T24	13,659	41,103	6,808	61,569	3,126	1,701	-191	4,637	8	0.09
				Avg	71,249	3,386	2,143	-458	5,071	8	0.10
				Min	49,528	2,750	1,701	-1,240	4,144	5	0.08
				Max	91,178	3,952	2,861	-132	5,769	11	0.11
		10 + 2.5 (Upstream + Downstream) Leaks									
3	Old	15,967	35,402	8,477	59,846	3,382	1,953	-1	5,334	10	0.10
	New	15,916	34,828	7,880	58,624	3,368	1,921	-1	5,288	10	0.10
	T24	13,606	30,467	5,039	49,113	2,881	1,668	-1	4,548	10	0.09
9	Old	15,953	61,399	13,943	91,295	3,480	3,290	-1	6,769	8	0.13
	New	16,124	59,437	17,963	93,523	3,479	3,012	-1	6,489	7	0.12
	T24	12,747	48,672	6,782	68,201	2,808	2,135	0	4,942	8	0.09
12	Old	17,665	51,120	14,284	83,070	3,814	1,965	-2	5,777	7	0.11
	New	17,413	49,302	13,013	79,728	3,751	1,836	-1	5,585	8	0.11
	T24	13,449	40,939	6,997	61,385	2,916	1,537	-1	4,452	8	0.08
				Avg	71,643	3,320	2,146	-1	5,465	8	0.10
				Min	49,113	2,808	1,537	-2	4,452	7	0.08
				Max	93,523	3,814	3,290	0	6,769	10	0.13

Table V-1b. Leakage Impacts on Annual HVAC System Operating Costs

		Annual Operating Cost [$]				Cost Increase Due to Leakage					
		Both				Fans	Cooling	Heating	Total		
CZ	Vintage	Fans	Cooling	Heating	Total	$/yr	$/yr	$/yr	$/yr	%	$/ft2
		2.5 + 10 (Upstream + Downstream) Leaks									
3	Old	14,598	34,682	8,089	57,369	2,014	1,233	-389	2,857	5	0.05
	New	14,577	34,116	7,436	56,129	2,028	1,209	-445	2,793	5	0.05
	T24	12,567	29,868	4,843	47,277	1,842	1,068	-197	2,713	6	0.05
9	Old	13,652	59,087	12,524	85,263	1,180	978	-1,421	737	1	0.01
	New	13,553	57,074	16,128	86,754	908	648	-1,836	-280	0	-0.01
	T24	11,156	47,481	6,178	64,815	1,217	944	-605	1,556	2	0.03
12	Old	15,826	50,291	13,853	79,970	1,975	1,135	-433	2,677	3	0.05
	New	15,660	48,535	12,510	76,704	1,998	1,068	-505	2,562	3	0.05
	T24	12,183	40,355	6,716	59,254	1,650	953	-282	2,321	4	0.04
				Avg	68,171	1,646	1,026	-679	1,993	3	0.04
				Min	47,277	908	648	-1,836	-280	0	-0.01
				Max	86,754	2,028	1,233	-197	2,857	6	0.05
		5 + 5 (Upstream + Downstream) Leaks									
3	Old	14,244	34,494	8,346	57,084	1,660	1,044	-131	2,573	5	0.05
	New	14,208	33,917	7,731	55,857	1,660	1,010	-149	2,520	5	0.05
	T24	12,182	29,678	4,973	46,833	1,457	879	-67	2,269	5	0.04
9	Old	13,874	59,428	13,452	86,754	1,401	1,318	-492	2,227	3	0.04
	New	13,952	57,581	17,336	88,869	1,307	1,155	-627	1,835	2	0.03
	T24	11,164	47,504	6,579	65,247	1,225	967	-203	1,988	3	0.04
12	Old	15,625	50,185	14,140	79,951	1,774	1,030	-146	2,658	3	0.05
	New	15,425	48,406	12,842	76,673	1,762	940	-172	2,530	3	0.05
	T24	11,936	40,227	6,901	59,065	1,404	826	-97	2,132	4	0.04
				Avg	68,481	1,516	1,019	-232	2,303	4	0.04
				Min	46,833	1,225	826	-627	1,835	2	0.03
				Max	88,869	1,774	1,318	-67	2,658	5	0.05
		2.5 + 2.5 (Upstream + Downstream) Leaks									
3	Old	12,585	33,450	8,478	54,512						
	New	12,549	32,907	7,881	53,336						
	T24	10,725	28,799	5,040	44,564						
9	Old	12,472	58,110	13,944	84,526						
	New	12,645	56,426	17,964	87,035						
	T24	9,939	46,537	6,783	63,259						
12	Old	13,852	49,156	14,285	77,293						
	New	13,662	47,466	13,014	74,143						
	T24	10,533	39,401	6,998	56,933						
				Avg	66,178						
				Min	44,564						
				Max	87,035						